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# U. S. S. *TENNESSEE*.

## DESCRIPTION AND OFFICIAL TRIALS.

BY LIEUT. COMDR. C. A. JONES, U. S. N. MEMBER.

The U. S. S. *Tennessee* was authorized by Act of Congress March 3, 1915, with a limit of cost of hull and machinery of $12,750,000. The *Tennessee* and *California* were the first two vessels authorized which embody the lesson of recent wars, pertaining to water tight integrity, and the use of electric propulsion allowed taking advantage of these lessons to the fullest extent.

The contracts for the main propelling machinery, its necessary auxiliaries, the six 300 K.W. turbo-generating sets and necessary spare parts were let to the Westinghouse Electric and Manufacturing Company May 12, 1916, for a total cost of $561,217.70.

The keel was laid May 14, 1917, and she was launched April 30, 1919, at the Navy Yard, New York. The *Tennessee* was commissioned on June 3, 1920, and since that date has completed successful final machinery and gunnery trials and seen several months' active service in the Fleet, where she has demonstrated her reliability and efficiency.

HULL DATA.

| | | |
|---|---|---|
| Draft designed, forward, feet and inches | | 29—11¼ |
| aft, feet and inches | | 30—6¾ |
| mean, feet and inches | | 30—3 |
| Displacement on designed draft, tons | | 32,300 |
| per inch on L. W. L., tons | | 101.1 |
| Length between perpendiculars, feet and inches | | 600—0 |
| Length over all, feet and inches | | 624—0 |
| Length on L. W. L., feet and inches | | 600—0 |
| Beam, extreme, feet and inches | | 97—5¾ |
| Beam on L. W. L., feet and inches | | 97—5¾ |
| Beam, moulded, feet and inches | | 96—0 |
| At designed draft | Area immersed midship section, sq. ft. | 2,875 |
| | Area L. W. L. plane, sq. ft. | 42,400 |
| | Wetted surface, sq. ft. | 75,750 |
| Coefficient of fineness, block | | .635 |
| midship section | | .977 |
| L. W. L. | | .726 |
| Fuel oil, 95 per cent capacity of compartments, tons | | 4,656 |
| Reserve feed water, capacity of compartments, tons | | 588 |

NOTE—The displacement includes two-thirds full supply of stores, fuel and water; full supply of ammunition.

GENERAL DESCRIPTION OF HULL.

In profile the *Tennessee* differs materially from that of any previous battleship. She has two smoke stacks, the after stack being surrounded with a structure for four 36-inch searchlights and control towers, the new standard bridge structure, and large control tops fitted on the usual two cage masts, thus giving her a more massive profile than in ships built heretofore. The after cage mast is also surrounded with platforms for 4 searchlights and gunnery control platforms. All of this can

be seen from the accompanying picture, Figure 1. The cage masts are provided with the usual wireless, signal yards, etc., except that the foremast has no topmast.

*Superstructure Deck and Bridges*—The superstructure deck extends from turret 11 to frame 87. On this deck are mounted four 5-inch guns, four 3-inch anti-aircraft guns, four 6-pounder saluting battery guns, boat stowage, garbage incinerator, boat cranes, etc. The standard bridge structure installed contains ten emergency staterooms, three baths and toilet (one on each of the three levels) ventilating trunks for forward engine room and forward four fire rooms, signal bridge, chart house, plotting room, radio room, navigating bridge and glass-enclosed pilot house.

*Upper Deck*—This deck extends from the bow, being a weather deck to frame 50, from which point it is covered by the superstructure deck to frame 87. There is a weather passageway on each side for the full length. Forward are the windlass, anchor chains, two electric deck winches, and turrets I and II. In the compartment covered by the superstructure deck is mounted a portion of the 5-inch battery in casements; crew's and officer's galleys; combined blacksmith, coppersmith, pipe and foundry shops; bakery and butcher shop.

*Main Deck*—The main deck is a weather deck abaft the break in the upper deck at frame 87. On this portion are turrets III and IV, two electric deck winches, an electric capstan, and supply and exhaust trunk openings for ventilation to the machinery spaces. The covered portion forward contains crew's living quarters, crew's reception room, postoffice, ship's store, armory, printing office, tailor shop, carpenter's and ship fitter's shop, etc.

*Second Deck*—Located on this deck are the crew's and officer's living quarters, sick bay and dental office.

*Third Deck*—On this deck are located crew's and officer's living quarters, storerooms, windlass machinery, electric-driven air compressors, submersible pump equipment with its motor generators, battle dressing station, general workshop,

electric repair shop, laundry, main radio rooms, kite ballroom winch, boat crane machinery, capstan machinery and cold storage rooms.

*First Platform, Second Platform and Hold*—On these decks are located machinery spaces, magazines and shell stowage, turret handling rooms, torpedo room, fuel oil stowage tanks, reserve feed bottoms, battery control rooms and store rooms.

## COMPLEMENT.

The ship's complement consists of the following:

| | |
|---|---|
| Commanding Officer | 1 |
| Wardroom, line officers (7 engineering) | 23 |
| staff officers | 4 |
| Junior officers, line | 27 |
| staff | 1 |
| Warrant officers (5 engineering) | 12 |
| Seaman branch | 739 |
| Artificer branch | 121 |
| engine room force | 263 |
| Commissary branch | 24 |
| Special branch, including yeoman, storekeepers, pharmacists, hospital apprentices, buglers and musicians | 78 |
| Messman branch | 41 |
| Total | 1,333 |

## BATTERY.

The main battery consists of twelve 14-inch, 50 caliber breech-loading rifles, mounted three abreast in four electrically-controlled balanced turrets. The turrets are on the fore and aft center line of the ship, two forward and two aft. The guns are mounted in individual slides and can be elevated independently.

The secondary battery consists of fourteen 5-inch, 51-caliber rapid-fire guns. Four are mounted on the superstructure deck, port and starboard. Ten are mounted on the inclosed portion of the upper deck, port and starboard.

The 5-inch battery is served by eighteen, 3-horsepower electric-driven chain ammunition hoists.

Additional guns consist of four 3-inch aero guns and four 6-pounder guns for saluting, located on the superstructure deck; two 1-pounder guns for boats, located in vent A-6; one 3-inch field piece stowed on the quarter deck, and 2 .30-caliber machine guns stowed in the armory.

The torpedo equipment consists of two 6.8 meters by 21 inches, submerged torpedo tubes mounted in the ship's side, just above the second platform, located between frames 16 and 17 starboard and between frames 21 and 22 port.

SMALL BOATS.

1—50-foot oil-burning steamer, reduction gear turbine driven.
2—35-foot motor boats.
2—50-foot motor sailing launches.
2—40-foot motor sailing launches.
1—31-foot racing cutter.
2—30-foot whale boats.
2—30-foot dinghies.
2—14-foot punts.

All boats are stowed on the superstructure deck, with the exception of the two 30-foot whaleboats, which are swung outboard on their davits at the upper deck level between frames 84 and 91, port and starboard.

BOAT CRANES.

For handling the boats there are two cranes of the pillar type installed on the superstructure deck. They are designed for a working load of 40,000 pounds. For handling lighter loads a whip is provided, which has a speed of 80 feet per minute at a working load of 10,000 pounds. Each boat crane has a hoisting motor of 50 horsepower and a training motor of 50 horsepower. Both motors are manufactured by the General Electric Company. The operating machinery for the boat cranes is located on the third deck.

### TOWING ARRANGEMENTS.

A 2½-inch diameter steel wire rope hawser, passed around the barbette of turret IV, including connecting shackles, etc., is provided for towing.

### ANCHOR HANDLING.

The windlass which was constructed by the American Engineering Company of Philadelphia, Pa., is of the vertical type, operated by two electric motors, and consists of three wildcats on vertical shafts, with locking gears, friction band brakes and bedplates. Each vertical shaft has a worm wheel on the lower end and is driven by a single worm on the extension shaft from the motors.

The electric motors are of 125 horsepower, 250 R.P.M., manufactured by the Diehl Manufacturing Company. The control is of the distant master controller contactor type and manufactured by the Cutler Hammer Mfg. Co.

### DECK WINCHES.

Four electric-driven, compound-geared deck winches are provided, two forward on the forecastle and two aft on the quarterdeck. They were manufactured by the Munsey Engineering Company and are driven by a 45-horsepower motor, manufactured by the Westinghouse Company.

### CAPSTAN.

An electric-driven capstan, manufactured by the Munsey Engineering Company, is located on the main deck, aft of turret IV. The machinery is located on the third deck, is compound geared and driven by a General Electric 45-horsepower motor.

### KITE BALLOON WINCH.

A kite balloon winch is installed aft on the third deck, the cable lead coming down through a protecting tube from the quarter deck. It is of the N.C.L. indirect spooling type and

designed to handle observation balloons through a wire at a considerable range of speeds and under widely fluctuating tension. The winch is driven by a General Electric 50-horsepower motor at 725 R.P.M.

STEERING GEAR.

The steering arrangement is an electro-hydraulic system having two main hydraulic plungers, each working on a pair of opposed hydraulic cylinders having connecting rods going to the rudder cross-head. The cross-head is keyed to the upper end of the rudder stock, the rudder being of the balanced type with an area of 456 square feet.

Oil is pumped from one side to the other of the main cylinders by two variable stroke pumps, each consisting of the pump end of a size 50, Type K Waterbury hydraulic speed gear. The two pumps which work together are directly connected by flexible couplings to the ends of the shaft of a 75-horsepower Westinghouse motor. A duplicate set of motor and pumps is provided as a reserve. Either set can be connected to the system by the operation of a three-way valve.

The amount of oil pumped and the direction of its flow is dependent on the movement in amount and direction of the tilting box of the pump. The tilting box is moved by a control shaft.

The control shaft is also connected to the follow-up mechanism which brings the pump tilting box back to neutral, thus stopping the movement of the rudder as soon as the controller is moved to the "off" position. A pressure operated control shaft stop is provided which limits the angle of tilt of the pump tilting box on high pressure, thus preventing overloading and stalling the motor.

The control shaft is operated by either one of a duplicate set of 1-horsepower pilot motors, magnetic brake, and control panel. This motor is started, stopped or reversed by a standard master controller of the nonfollow-up type, located at the

various steering stations. The control shaft can also be operated manually by a "trick" wheel from the steering room.

Provision is made for hand steering and auxiliary drives. The system consists of a hydraulic pump of the same size and construction as those used by the power drive, arranged to be driven by any one of the following methods: (1) handwheel; (2) 7½-horsepower auxiliary motor, supplied with power by batteries; (3) 5-horsepower compressed air motor. Only local hand control is provided.

## LAUNDRY.

The laundry is located on the third deck and equipped with the following apparatus:

2 Universal presses, 88 inches.
1 Combination ironer.
1 Flat work ironer, return apron type.
1 Collar shaper.
2 Extractors, 26-inch.
1 Drying room, 5-drawer.
1 Tumbler dryer.
1 Disinfector.
1 Washer, 41x72 inches.
1 Ironing table.
1 Soap tank.
1 Starch kettle.
1 2-Compartment tub.
3 Truck tubs.

## GALLEYS.

Ten oil-burning Navy standard ranges are installed in the crew's galley and four in the officer's galley. The primary system of oil burning is by air atomization, but steam can be used in an emergency. Oil tanks connected to the booster pump discharge are provided. The oil is forced to the burners from these tanks by air pressure applied to the top of the oil.

A separate driven air compressor, manufactured by the National Brake and Electric Company, is installed for supplying air to the ranges and incinerator at 35 pounds pressure.

The following additional equipment is installed in the crew's galley:

4 Steam-jacketed copper kettles, 80 gallons each.
2 Steam-jacketed copper kettles, 60-gallons each.
3 Coffee urns, 80 gallons each.
1 Coffee urn, 20 gallons.
1 Meat grinder, electric driven.
1 Kitchen and cake mixer, 80 quarts, electric driven.
1 Meat slicer, hand-operated.
1 Potato peeling machine, electric driven, located in potato peeler room.
1 Dish washing machine, electric driven, located in general mess pantry.
4 Warming ovens located in serving room.
1 Butter slicer, electric driven, located in general mess issuing room.
1 Ice cream freezer, 40 quarts, located on second deck.

BAKERY.

The following equipment is installed in the bakery:

2 Bake ovens, electric, No. 1 U. S. Navy Standard.
1 Steam box.
2 Dough troughs, portable.
1 Dough mixing machine, electric driven.
1 Kitchen and cake mixing machine, electric driven.

INCINERATOR.

A garbage and refuse incinerator of the Jockers type, which was manufactured at the Navy Yard, New York, is installed on the starboard side of the superstructure deck. The oil supply is taken from the 150-gallon fuel oil supply tank located on the main deck. Air is supplied by the galley compressor. A ¼-h.p. electric-driven blower is used to supply forced draft.

## DRAINAGE SYSTEM.

The drainage system in general consists of a main drain, secondary drain forward, and secondary drain aft. The main drain has no connection or by-pass to the forward secondary drain but a 6-inch line connects the main and after secondary drain.

The main drain consists of four independent units, two in each machinery room. One of these is connected to the main circulating pump and one 8-inch main to the fire and bilge pumps. The connection to the main circulating pump consists of a single suction, the valve being interlocked with the sea suction to prevent opening both at the same time.

The forward secondary drain is connected to a direct-connected electric-driven centrifugal pump located in compartment "A-109." This pump is capable of discharging 750 G.P.M. against a head equivalent to a pressure of 35 pounds per square inch. The after secondary drain is connected with the shaft alley bilge pump of the electric-driven plunger type, located in compartment "D-102." This pump is capable of delivering 250 G.P.M. against a head of 30 feet.

There are also two portable pumps for draining compartments not permanently connected to the drainage system.

## SUBMERSIBLE PUMP SYSTEM.

Two submersible salvage pumps of the alternating current type are furnished. The plans were furnished by Submersible and J. L. Motors of Middlesex, England, and built by the Triumph Electric Company.

One unit consists of the following:

15 horsepower, 220 volts, 3 phase, 60 cycles, 1,800 R.P.M., alternating current motor, direct connected to a 4-inch centrifugal pump rated at 430 gallons per minute at 75-foot head.

1—Motor Compensator, alternating current.

2—Special 3-conductor cables, each 75 feet long.

2—Reels for cable.
1—Foot valve and strainer, 4-inch.
3—Suction Hose, 4-inch, each 8-foot long.
1—Suction elbow.
2—Canvas Discharge Hose, 4-inch, each 50 feet long.
1—Wooden Truck, with rollers.

Water is circulated through the motor windings for cooling purposes, special rubber insulation being used in the stator windings.

Trouble has been experienced with this equipment, but a new design is now being perfected. This design will use direct current and be of smaller size.

Terminals for connecting the pump are distributed throughout the ship as necessary, all being located behind armor. Power is supplied by two 50 K.V.A. motor generating sets, located in the air compressor rooms on the third deck.

### FIRE SYSTEM.

The fire main is designed so that the loop in the machinery room, boiler rooms, and store rooms below the third deck, are thoroughly protected. There are two risers in the barbettes, one in barbette II and one in barbette III. The risers in the barbettes have branches at each deck, fitted with cutout valves, operated from outside the barbettes and just above hose connections.

### FIRE EXTINGUISHERS.

There is installed in each of the fire rooms and machinery rooms, for extinguishing oil fires, one extinguisher of the foam type together with the necessary hose. Two solutions are employed which, when combined, generate carbon dioxide gas and forms a heavy foam. This foam, by blanketing the burning surface, extinguishes the fire. Portable Pyrene (carbon tetrachloride) extinguishers are located in various parts of the ship.

### MAGAZINE FLOODING SYSTEM.

Provision is made for flooding all magazines by use of sprinkling pipes or flooding direct through valves.

### SANITARY SYSTEM.

The sanitary system with exception of the independent flushing system for crew's, petty officer's and mess attendant's toilets, is supplied from the fire main.

For the independent system there are located in compartment A-109, two electric-driven centrifugal pumps connected to a 9-inch sea valve. Each of the pumps have a 5-inch suction from the sea and a 5-inch discharge, capable of delivering 500 G.P.M. against a total head of 50 feet. For emergency, connections are made to the forward secondary drainage pump and the fire main.

### FRESH WATER SYSTEM.

The main fresh water tanks are located in compartments A-114, A-115 and A-116. Their capacities in gallons are as follows: A-114, 23,970; A-115 and A-116, 14,790 each; total, 53,550 gallons. They are filled by a 2½-inch distiller main from the evaporator room, which also has connections on the ship's sides for obtaining water from an outside source. A separate tank is provided for the crew's amidships and fireman's washrooms of 300 gallons capacity, direct connected to the distilling plant as well as the fresh water delivery main.

In pump room A-109 are located two direct connected electric-driven centrifugal pumps, each capable of delivering 200 gallons, against a pressure of 35 pounds. These pumps take their suction from the fresh water tanks and by suitable piping deliver water to outlets throughout the ship. Lines serving different systems are fitted with recording meters in order to keep a close check on water used for different purposes.

### COMPRESSED AIR FOR TORPEDOES.

Two Mark VII, model 5, electrically-driven, 4 stage Pratt Iron Works Company's torpedo air compressors are installed, one in torpedo room A-202, and one in after torpedo air compressor room D-227. Each compressor has a capacity of 50 cubic feet per hour at 2,500 pounds pressure per square inch and is supplied with circulating water from a 2-inch sea chest or from the fire main.

Two Mark VII Model I accumulators each consisting of two sections composed of 16 flasks of individual capacity of 0.52 cubic feet, are fitted, one in torpedo room A-202 and one in air compressor room D-227.

### COMPRESSED AIR AND GAS EJECTING SYSTEMS.

Compressed air for the gas ejecting system is supplied from four sources as follows: Two compressors in compressor rooms A-509, two in D-501, the accumulator in the after torpedo air compressor room D-227, and the steam-driven compressors in the engine rooms. The compressors on the third deck are of the Blake and Knowles two-stage vertical type, geared, electric driven, and with individual capacities of 175 cubic feet of free air per minute at 150 pounds per square inch pressure. They are fitted with reservoir tanks and arranged for automatic starting and stopping. Automatic unloading valves for operating on stopping are fitted in order to prevent stalling when starting.

Any one of the four compressors described above or the two Westinghouse steam-driven compressors, one located in each engine room, with individual capacities of 45 cubic feet of free air per minute at 150 pounds pressure, may be used for supplying air to the foundry, forge, pneumatic tools, pneumatic-tube system, operation of air motor of steering engine, etc. Each of the Westinghouse compressors have 11 by 11-inch air cylinders by 12-inch stroke and is connected to a reservoir tank.

## VENTILATION AND HEATING SYSTEMS.

To provide ventilation throughout the ship 79 supply and exhaust ventilating sets have been installed. Each set, consisting of a centrifugal multivane type fan driven by an electric motor, is connected to a system of ducts from which branches are led to the compartments requiring artificial ventilation.

In living quarters, crew's spaces, etc., where heating is required, the air is passed through steam coil heaters before delivery to the ducts. The heaters are arranged so that either one, two or three coils may be used; the temperature can also be regulated by operating a by-pass control damper. The humidity is controlled automatically by means of a hygrostat, located in typical compartment, supplied by the heater. The hygrostat operates a valve admitting steam direct into the ventilating air. The system is designed to maintain a temperature of 70 degrees Fahrenheit, with the air outside at 0 degrees Fahrenheit. Some heating is done by use of steam radiators and also by use of electric heaters.

## MAIN PROPELLING MACHINERY.

The main propelling machinery of the *Tennessee* consists of two turbine-driven alternators, two main condensers, four induction motors, two booster generator sets for excitation supplied with power by any one of six 300-K.W. geared turbo-generating sets, motor and steam-driven auxiliaries, main switching equipment, exciter and auxiliary switchboard, ventilating blowers for main motors, control panels and wiring for electric auxiliaries, and necessary wiring for connecting up main generators and motors with the switching equipment.

The weight of the main propelling machinery, as listed above, is about 1,700 tons.

The general arrangement of the machinery is shown in Figure 2. In all firerooms, lower engine rooms, evaporator

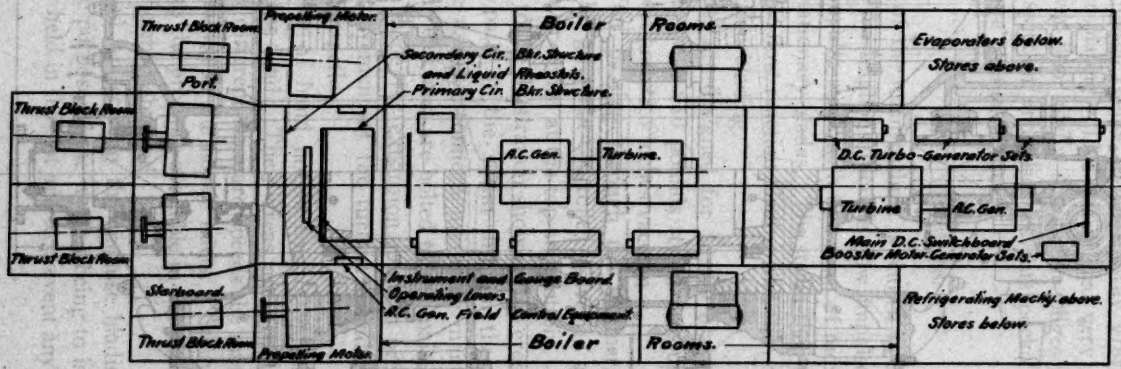



FIG. 2.—GENERAL ARRANGEMENT OF MACHINERY COMPARTMENTS.

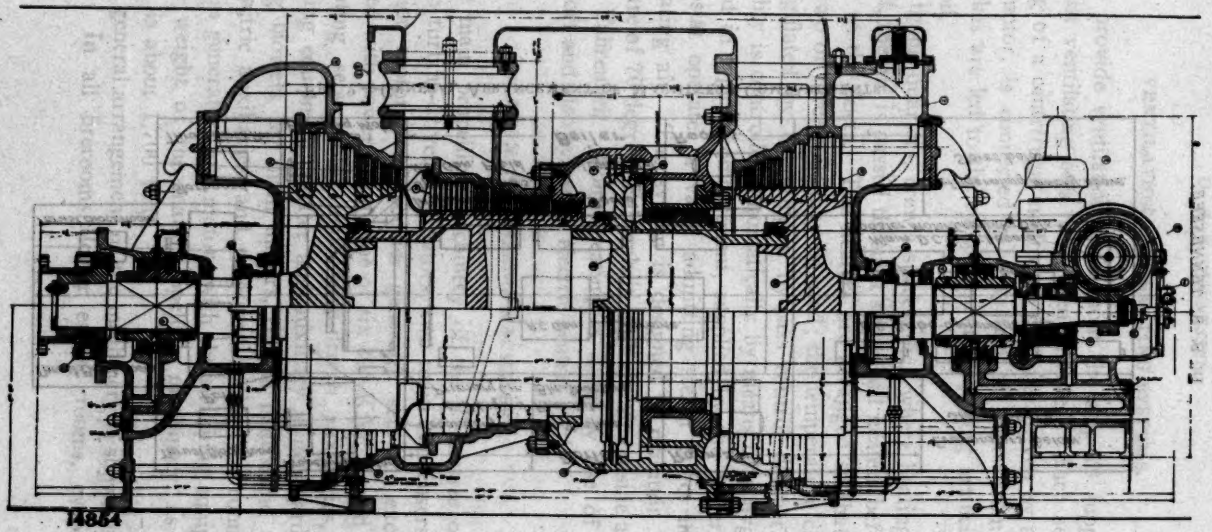



FIG. 3.—LONGITUDINAL SECTION OF MAIN TURBINE.

rooms, and motor rooms Irving subway grating is used instead of floor plating. This has proved very successful. Its advantage over floor plates is that the bilges are open at all times for inspection and also prevents pockets of gas accumulating where oil fuel is used.

## MAIN TURBINES.

A section of the main turbine is shown in Figure 3. The turbine is a Westinghouse double flow impulse reaction type, having one impulse stage with two rows of moving blades and one row of stationary blades. The steam then flows through a series of single flow reaction blading and then a series of double flow reaction blading.

The turbines are designed for 16,000 brake horsepower at 2,075 R.P.M., with 250 pounds steam pressure gage, 50 degrees superheat and 28.5 inches vacuum.

Steam, after passing through the strainer, main throttle valve, and the governor-controlled valve, is admitted to the turbine in the openings at the top and bottom.

Referring to the illustration showing the cross section of the turbine, the path of the steam through the turbine can be traced. The exhaust connections to the main condenser are shown at each end, and are bolted to the main condenser, provision being made for expansion joint to prevent restriction of the end movement of the turbine cylinder caused by expansion or contraction.

The number of hand nozzle control valves to be opened depends on the load. Their areas are arranged so that by proper use at different standard speeds the maximum pressure can be maintained at the nozzle inlets. The nearer this pressure is kept to the pressure available in the main steam line the better the efficiency obtained.

The method of securing the impulse blading is shown in Figure 4, the purpose of the wedges being to insure the blades being tightly held in the slot and to prevent any vibration being

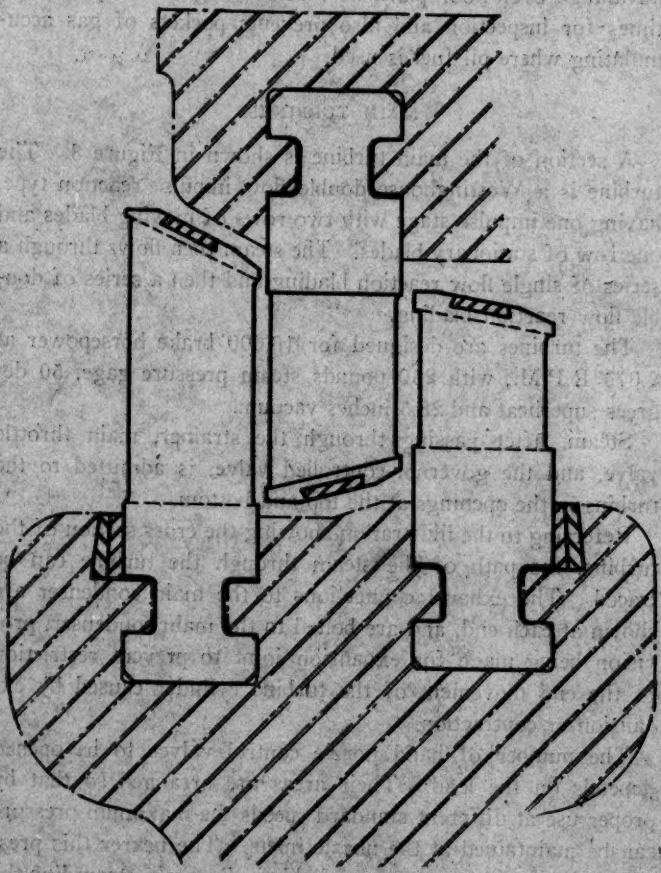


FIG. 4.—METHOD OF SECURING IMPULSE BLADING.

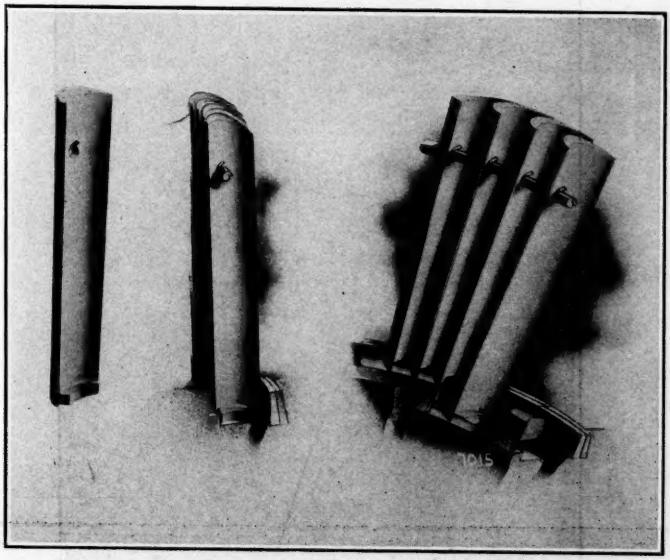


FIG. 5.—METHOD OF SECURING REACTION BLADING.

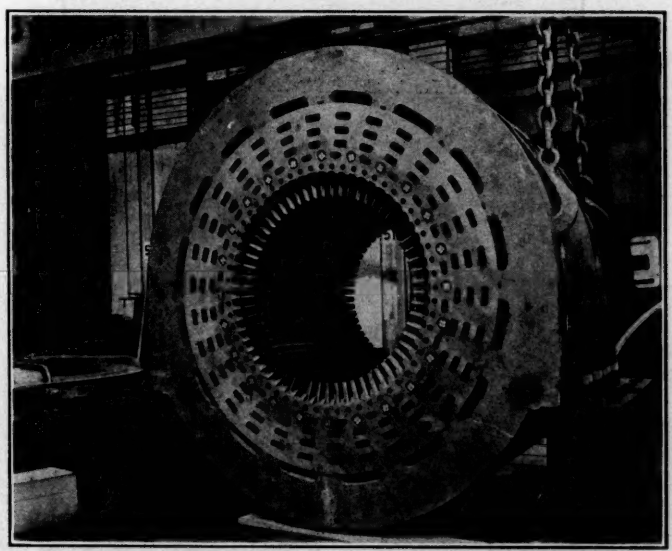


FIG. 10.—MAIN GENERATOR STATOR READY FOR WINDING.

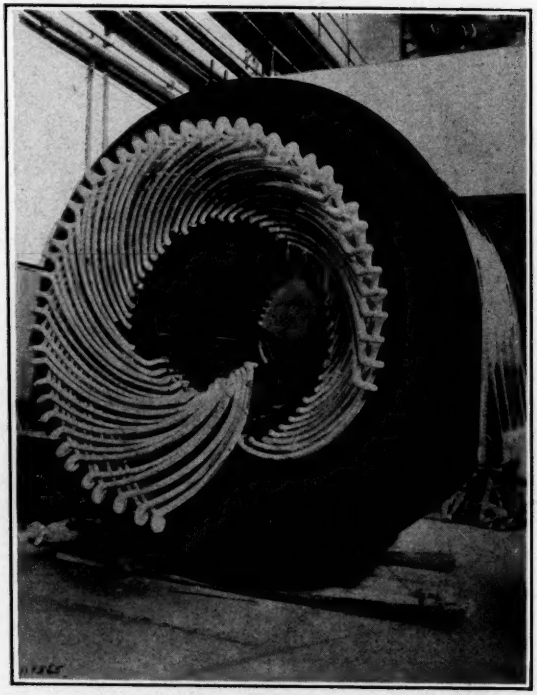


FIG. 11.—PROCESS OF WINDING MAIN GENERATOR STATOR.

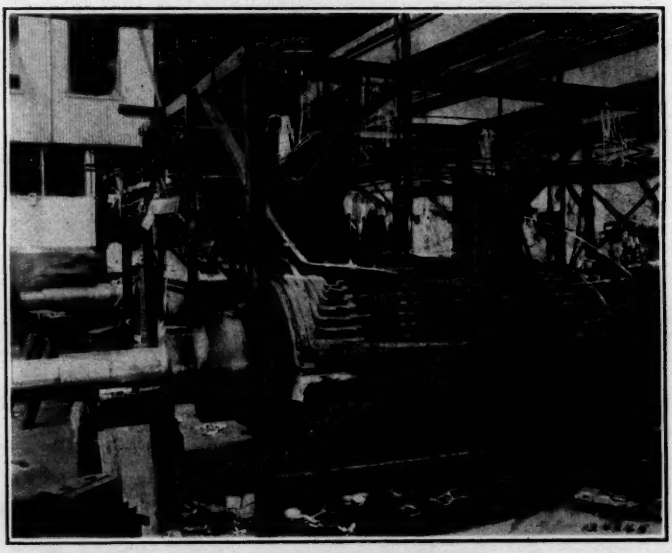


FIG. 12.—PROCESS OF WINDING MAIN GENERATOR ROTOR.

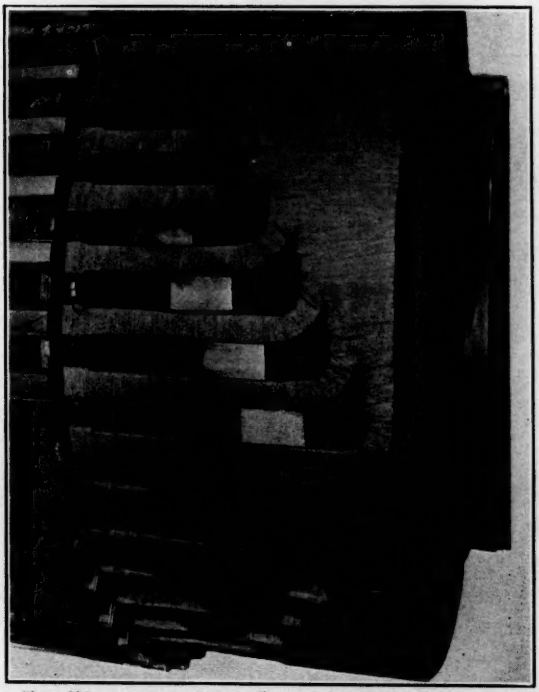


FIG. 13.—END WINDINGS OF MAIN GENERATOR ROTOR SHOWING BLOCKING.

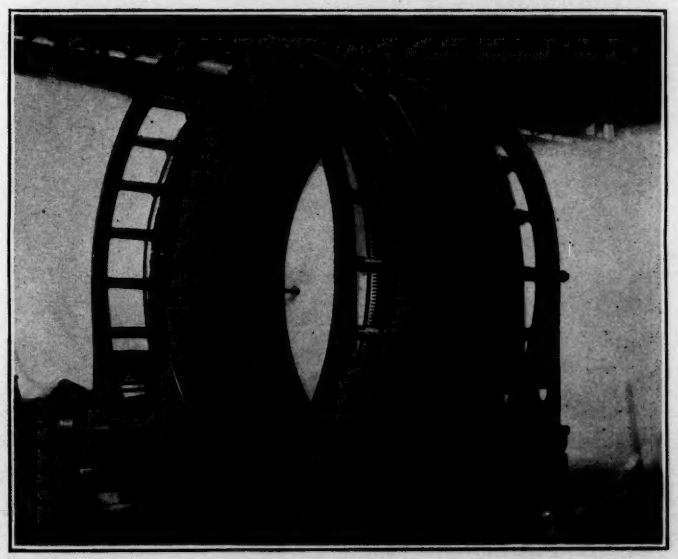


FIG. 16.—MAIN PROPELLING MOTOR STATOR READY FOR WINDING.

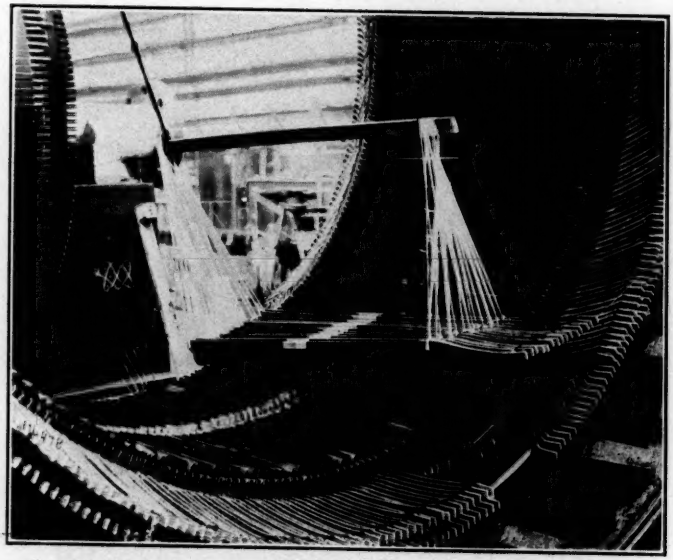


FIG. 17.—MAIN PROPELLING MOTOR, LAYING THE 24 POLE WINDINGS.

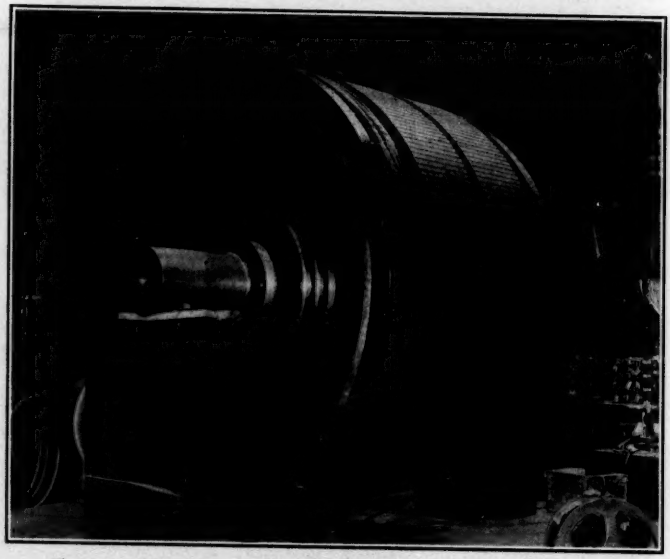


FIG. 18.—MAIN PROPELLING MOTOR ROTOR, WINDING COMPLETED.

conveyed to the reduced section. The method of attaching the reaction blading is shown in Figure 5. One of the important improvements of these reaction blades is the tapering of the blade section so that in the event of blade vibration the stresses will be distributed over a considerable length of the blade instead of being concentrated at the point of attachment. This thickening is not clearly shown in this illustration. It can be seen from the system used to hold the blade and the interlocking, that no caulking is necessary. Lacing wires, the number depending on the dimensions of the blades, are provided for bracing the upper ends of the blades.

The steam thrust, due to the single flow reaction blading is partially balanced by a balance piston or dummy. The dummy piston is formed by a forged steel ring shrunk and keyed on the turbine spindle. This ring and the turbine cylinder opposite are fitted with suitably shaped strips of soft brass, having a thin edge which limits the leakage of steam from the high pressure to the low pressure side of the dummy. The remaining thrust is taken by a Kingsbury thrust bearing. The thrust bearing shoes and base rings are supported in a cage. Liners are placed between the cage and its cover so that the working clearance between the babitted faces of the shoes and the thrust collar can be adjusted. By suitable gearing the cage can be moved axially from a connection on the outside of the thrust bearing pedestal to adjust the endwise clearance. This clearance is set with the turbine cylinder hot and while rotating slowly and should be about 1/32-inch on the side towards the thrust bearing. A scale at the connection on the thrust bearing pedestal indicates the amount the spindle is moved.

The coupling used for driving the generator permits the generator rotor to move a limited amount in an axial direction in order to obtain flexibility. The type of coupling used is a pin type, coupling sleeves being pressed and keyed on the turbine spindle and generator driving shafts.

Glands of the combined steam and water sealed type are used to prevent leakage where the rotor shaft passes through the casing. The water gland is used at all speeds from and above half speed and below this speed the steam gland is used. The water gland operates on the principle of a centrifugal pump. The steam gland consists of a section of labyrinth packing and is supplied with steam at low pressure through reducing valves and control valve. Both the gland casing and the gland runners are made in halves to facilitate removal for replacement or overhaul.

To control the supply of steam and water to the glands a special automatic valve is fitted. This valve is controlled by a separate centrifugal governor, driven by the main turbine through a suitable set of gearing, which automatically shifts from steam to water or vice-versa, when the turbine reaches the correct speed.

Figure 6 shows the relation of the various parts of the turbine control system. In general the speed control consists of a centrifugal speed control governor driven from the turbine shaft by suitable gearing, the centrifugal action being balanced against an oil pressure which is varied by the setting of a speed control lever located in the control room. The speed control stand is supplied with a constant oil pressure. By a system of floating levers this pressure is reduced to any desired amount by balancing against a spring the compression of which is varied by the different settings of the speed control wheel. A double seated poppet valve, located on the main steam inlet to the turbine, is controlled through a floating lever oil pressure relay system from the main governor. This valve controls the amount of steam admitted to the turbine to maintain the speed in accordance with the setting of the speed control lever in the control room. A "vibrator" driven by a small motor on the speed control stand introduces a constant rapid and small pulsation in the oil pressure to the governor which causes a corresponding movement of all parts of the gov-

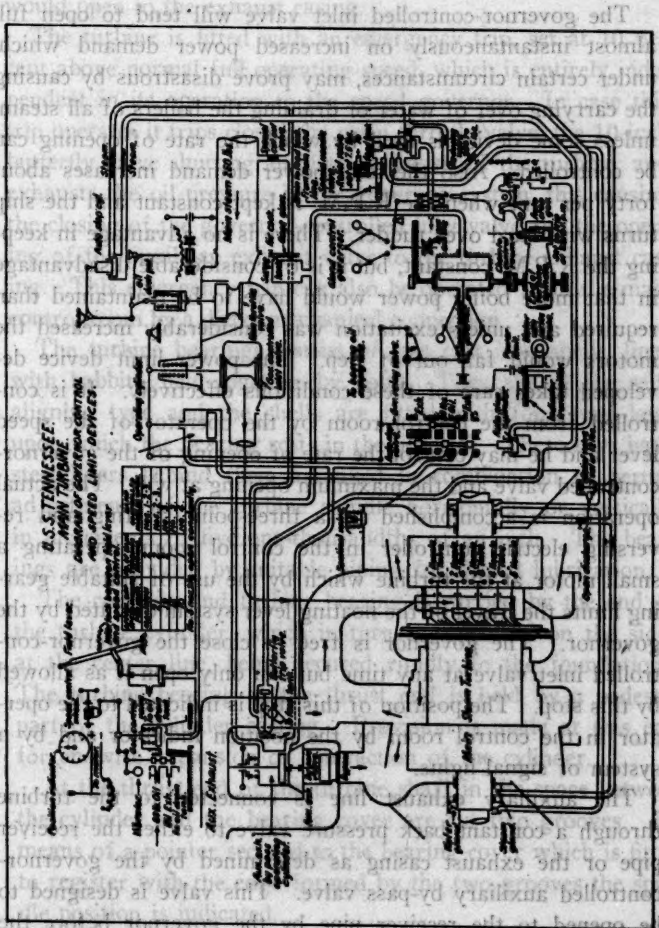



FIG. 6.—DIAGRAM OF TURBINE CONTROL SYSTEM.

ernor system. This prevents sticking, thus insuring rapid and accurate answer to any change.

The governor-controlled inlet valve will tend to open full almost instantaneously on increased power demand which, under certain circumstances, may prove disastrous by causing the carrying over of water or draining the boilers of all steam, unless some device is used by which the rate of opening can be controlled. Also the total power demand increases about forty per cent when the R.P.M. is kept constant and the ship turns with hard over rudder. There is no advantage in keeping the R.P.M. constant, but it is of considerable disadvantage in that more boiler power would have to be maintained than required and unless excitation was considerably increased the motors would fall out of step. The power limit device developed takes care of these conditions effectively. It is controlled from the control room by the operator of the speed lever and he may control the rate of opening of the governor-controlled valve and the maximum opening at will. The actual operation is accomplished by a three-point starting and reversing electric controller in the control room, operating a small motor at the turbine which by the use of suitable gearing limits the travel of the floating lever system operated by the governor. The governor is free to close the governor-controlled inlet valve at any time but can only open it as allowed by this stop. The position of this stop is indicated to the operator in the control room by the position indicator and by a system of signal lights.

The auxiliary exhaust line is connected to the turbine through a constant back pressure valve to either the receiver pipe or the exhaust casing as determined by the governor-controlled auxiliary by-pass valve. This valve is designed to be opened to the receiver pipe by the governor before the governor-controlled inlet valve opens and to close after the governor-controlled inlet valve closes in case the speed still exceeds the setting on the speed lever. In case of failure of oil

pressure, the turbine is protected as the governor-controlled inlet valve would close and the auxiliary exhaust by-pass valve would open to the exhaust casing.

The turbine is fitted with an emergency trip, set at 10 per cent above normal full operating speed, which is entirely independent in its operation to the speed governor. In case this trip operates it trips closed the main throttle valve, the 10-inch butterfly valve shutting auxiliary exhaust off the turbine, and exhausts the oil pressure in the governor system, this causing the closing of the governor-controlled inlet valve and the opening of the auxiliary exhaust valve to the turbine exhaust casing. This emergency trip can also be operated from the main control room by a direct mechanical connection.

The turbine bearings consist of two cast iron shells, lined with babbitt, held together by bolts. They are of the self-aligning type and the shells are fitted with four steel keys upon which the bearing rests in the pedestal. These keys have steel liners behind them arranged in combination to permit adjustments of the turbine spindle horizontally or vertically in increments of five one-thousandths of an inch. The bearings are provided by suitable piping for forced lubrication.

The coupling end turbine bearing is carried by the end of the turbine cylinder, which in turn is supported on the sides at the center line, being secured rigidly to the foundations. The turbine bearing at the thrust end is held by a pedestal part of the cylinder casting. Provision is made at this end for endwise expansion or contraction of the cylinder.

At the thrust end of the turbine shaft in the space between the cylinder and the bearing cover are cut two grooves. By means of a pointer secured to the bearing cover which is fitted to register with the edge formed by the two grooves the spindle position is indicated.

A barring over gear is fitted on the coupling between the turbine and generator for rotating the turbine and generator rotor.

A bridge gauge is furnished for checking the position of the turbine shaft relative to the turbine casing.

## MAIN CONDENSING APPARATUS.

*Main Condensers*—There are two main condensers, one for each main turbine. They are installed directly underneath the turbine and supported on foundations carried on the inner bottom. There is a copper expansion joint fitted between the turbine and condenser to allow for expansion and contraction.

The condensers are of the Lovekin type which he terms the "Circumflow" condenser. A cross section of this condenser is shown in Figure 7. It consists essentially of a primary and

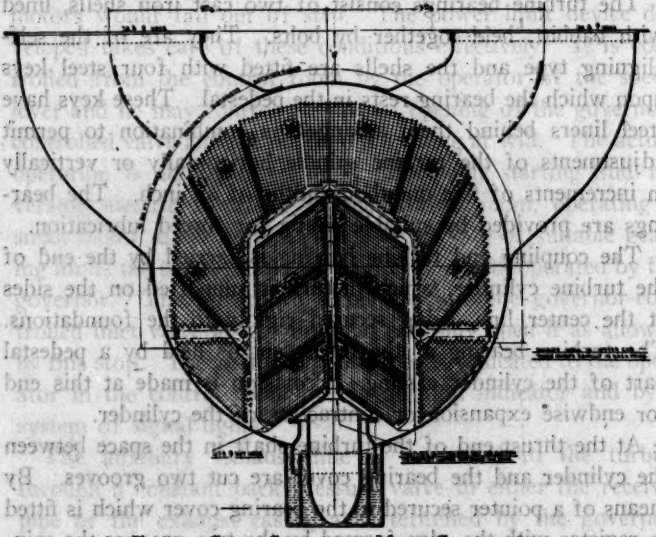


FIG. 7.—CROSS SECTION OF MAIN CONDENSER.

secondary condenser. The tubes are divided into subdivisions by means of so-called rain plates, serving to direct the steam flow and at the same time preventing an excessive amount of condensate from falling on the tube nests below.

The principal function of the secondary condenser is to cool the air, thus reducing its volume, and to prevent the formation of air pockets. The air is collected in the center channel and removed by the use of air pump or air ejectors. The division plate in the condenser head is arranged so that the major part of the circulating water first passes through the tubes of the secondary condenser and then through the tubes of the primary condenser. The advantages claimed due to the use of this condenser is more efficient heat transfer, thus reducing the weight, hotter condensate, no air pockets, and all parts of the condenser active. A very complete test was run on the condenser under representatives of the U. S. Naval Engineering Experiment Station, the results of which are not available at this time.

The shell of the condenser is of steel plate, rolled to shape and with only sufficient rivets to hold for electric welding. No trouble whatever has been experienced due to welding, and it resulted in a much cheaper and quicker job than riveting. This is indicated by the fact that the condenser went on the test stand and came off tight the same day after the hydrostatic test.

The tubes are straight, rolled into the tube sheets at one end and gland packed at the other end. Test for tightness is made on board ship by filling the condenser with water over the expansion joint connection, after sealing the turbine glands, and then applying air pressure to the turbine casing. Special clamps are furnished for supporting the expansion joints during this test.

PRINCIPAL DIMENSIONS.

| | |
|---|---|
| Diameter, inside, feet and inches | 9 — 0 |
| Thickness of shell, inch | 7/16 |
| Length between tube sheets, feet and inches | 10 — 9 |
| Thickness of tube sheets, inch | 1 1/4 |
| Tubes, number | 6,604 |
| diameter, outside, inch | 5/8 |
| thickness, inch | .065 |

| | |
|---|---|
| Cooling surface, square feet | 11,616 |
| Size of each main exhaust nozzle, square feet | 25.2 |
| Diameter, air pump suction, inches | 11 1/2 |
| condensate pump suction, inches | 10 |
| air ejector suction, each, inches | 5 |
| circulating water inlet and outlet, inches | 30 |

*Main Air Pumps*—One independent double, vertical, single acting air cylinder, and inverted double acting steam cylinder main air pump is installed for each main condenser.

PUMP DATA.

Make of Pump—M. T. Davidson Company.

| | |
|---|---|
| Diameter of air cylinders, inches | 34 |
| steam cylinders, inches | 13 |
| Stroke, inches | 27 |
| Suction opening, inches | 11½ |
| Discharge opening, inches | 10 |
| Steam connection, inches | 2 |
| Exhaust connection, inches | 2½ |
| Connection on suction pipe to reserve feed manifold, inches | 4½ |

*Main Condensate Pumps*—One motor-driven double inlet, volute, vertical, single stage and single runner for each condenser. The pump suction is taken off the main air pump suction pipe and discharge is to the air separating tank. The complete unit was manufactured by the Westinghouse Electric and Manufacturing Company.

PUMP DATA.

| | |
|---|---|
| Diameter, suction pipe, inches | 10 |
| discharge pipe, inches | 6 |
| Capacity, gallons per minute | 500 |
| Head, feet | 70 |
| Motor, horsepower | 19 |
| voltage | 240 |
| speed, R. P. M. | 1,400/1,700 |

windings. Compound with commutating poles.

Controller, type. Hand-operated sliding arm starting and field regulating rheostat with circuit breaker.

*Main Air Ejectors*—One three-unit set of Westinghouse Le Blane air ejectors, each of two stages in series, is installed for each main condenser.

EJECTOR DATA.

| | |
|---|---|
| Manufacturer—Westinghouse Electric & Manufacturing Co. | |
| Type | "G" |
| Steam pressure, pounds per square inch | 125 |
| Pounds of free air per hour at 28 inches vacuum referred to 30-inch barometer | 36 |
| Pounds of free air per hour at 28½ inches vacuum referred to 30-inch barometer | 27 |
| Steam consumption, pounds per hour | 880 |

The air ejectors discharge into air separating tanks for condensing the steam and for ridding the condensate of air. The major part of the steam is condensed by the discharge water from the condensate pump, thus regaining the latent heat of the steam used in the ejectors. The upper part of the air separating tanks, as shown in Figure 8 has packed tubes through which circulating water is passed.

*Air Separating Tank*—The details of the air separating tank can be seen in Figure 8.

Manufacturer—Westinghouse Electric and Manufacturing Co.

| | |
|---|---|
| Capacity, pounds of steam per hour | 4,400 |
| pounds of condensate per hour | 195,000 |
| Cooling surface, square feet | 90 |
| Cooling water required, gallons per minute | 200 |
| Resistance head, feet | 8 |
| Tubes, number | 326 |
| diameter, inch | ⅝ |
| length, inches | 22¾ |

*Main Circulating Pump*—There is one motor driven, variable speed, double inlet, horizontal, volute, single stage and double runner main circulating pump for each main condenser.

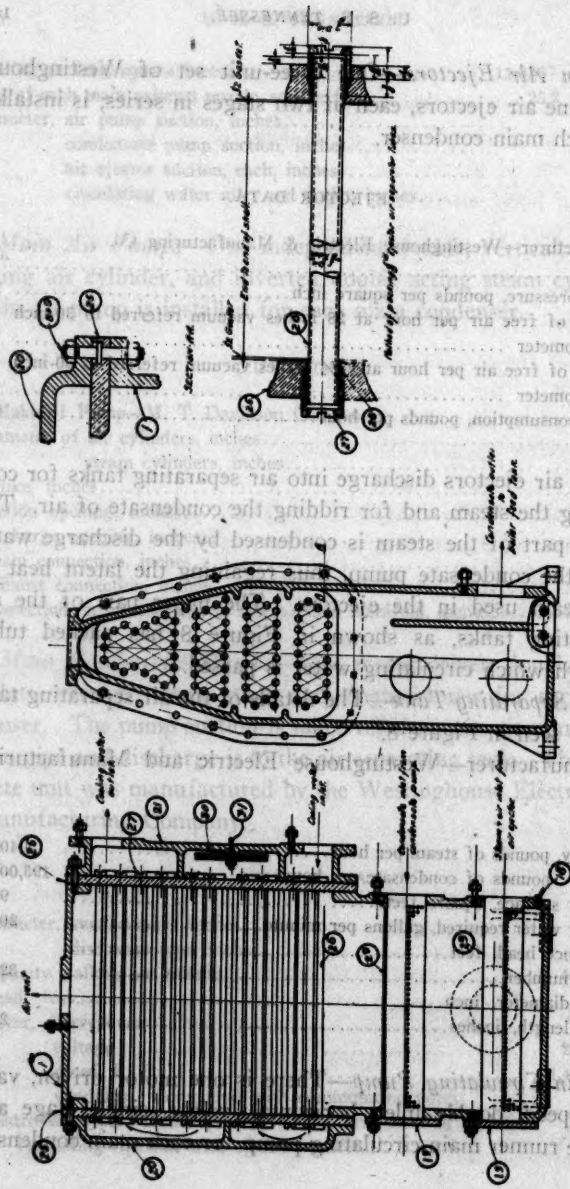


FIG 8.—AIR SEPARATING TANK.

### PUMP DATA.

The complete unit was manufactured by the Westinghouse Electric and Manufacturing Company.

| | |
|---|---|
| Capacity, G.P.M. at 700 R.P.M. | 19,000 |
| Head, feet, at 700 R.P.M. | 30 |
| Suction, openings, number | 2 |
| diameter, each, inches | 30 |
| drainage, inches | 15½ |
| Discharge, diameter, inches | 28 |
| oil cooler and water service, inches | 5 |
| Motor, windings. Compound with commutating poles. | |
| horsepower | 235 |
| voltage | 240 |
| speed, R.P.M. | 350/700 |
| Controller, type. Panel type of multiple lever starter, with circuit breakers fitted with overload coil and no voltage release, and field rheostat for speed control. | |

### MAIN GENERATORS.

Each generator has a nominal rating of 13,250 K.V.A. They are three phase, 34.6 cycles, 3,270 volts at 83.4 per cent power factor, and a maximum rating of 14,850 K.V.A. at 2,195 R.P.M. It is necessary that the field of generators designed for ship propulsion be designed with a good overload margin in order to take care of short time over excitation periods. Over excitation is necessary during starting and reversing in order to supply the necessary torque.

Here, as in all electrical apparatus for marine use, special provision must be made to prevent damage from salt contained in the air. This is accomplished in the main by the use of mica for insulating, only sufficient insulating material of other types being used as found necessary for the proper application of the mica. Special care is taken in applying and baking the several coats of moisture resisting varnish to seal the coils against salt moisture. The windings consists of two conductors per slot, each conductor being made up of a number of strands, individually insulated with mica tape to break up eddy currents.

Ventilation is provided by fans mounted on each end of the rotor. The air is drawn through an opening in the bottom of the end bells from a screened duct below the generator. In order to prevent excessive air noises from the fan being communicated to the engine room, the intake ducts are baffled for changing the direction of air flow. The fan, by building up a high air pressure in the end bells, forces this cooling air through the air-gap and axial ducts in the stator core to the central radial discharge openings and thence to the discharge duct.

The rotor is made of one piece steel forging provided with radial slots which carry the coils. Only mica or other fireproof material is used in insulating the rotor windings. Trouble was at first experienced by the end windings shifting position, but this, as demonstrated by severe shop tests and several months' actual service operation, has been overcome by proper blocking. No provision is made for ventilation of the rotor by circulating air in order to prevent the accumulation of salt deposit.

The generator is provided with steam connection for extinguishing fire. The supply valve is of special type so that in case of leakage by the valve the leakage passes to an open drain. There are also provided inlet and outlet dampers to the air supply and exhaust which must be closed when the machine is idle or in case of fire.

Steam heating coils are provided to keep the temperature of the generator slightly above the surrounding air in order to prevent sweating. All joints for the steam connections to the machine are made outside the casing.

The stator windings are provided with six thermo-couples, properly distributed, to measure the hot spot temperatures. The instrument for measuring the temperature is located in the control room.

The rotor is provided with pedestal bearings at each end. The bearing shells are of the self-aligning type, fitted for

forced lubrication. Liners are provided for accurately setting the air gap. The pedestal on the collector end is insulated from the foundation to prevent circulating stray field currents.

Figures 9 to 13 illustrate the construction of the main generator stator and rotor.

Excitation for the generator is provided from any one of the six 300 K.W. direct current turbo-generating sets installed, three in each engine room. The power required for excitation at full power is about 50 K.W. for each generator. Booster generating sets are provided for varying the excitation.

## MAIN MOTORS AND LIQUID RHEOSTATS.

The main motors are three phase, 24 pole, and have a rating of 7,000 H.P. at 3,270 volts and 170 R.P.M. Up to 15 knots they were designed to operate on 36 poles and have a rating of 2,125 H.P. at 3,250 volts and 118 R.P.M. A maximum rating of 8,375 H.P. at 180 R.P.M. for 4 hours is provided.

The details of construction can be seen in Figures 14 to 19. They are of the two-bearing bracket type, the rotor being supported by bearings carried in brackets which fit into recesses in the frame and bolted thereto. The frame consists of a single steel casting of skeleton structure. Inwardly projecting ribs are provided with slots for receiving the dovetailed segmental laminations. The supporting feet are cast integral with the frame, and extend across the entire length. Both sides of the laminations are coated with enamel in order to effectively break up the eddy currents. Finger plates and heavy through bolts are used to keep the laminations in place.

The rotor spider is a steel casting of the double arm construction, pressed on and keyed to the shaft. Slots are provided in projecting ribs for receiving the corresponding dovetails of the segmental laminations.

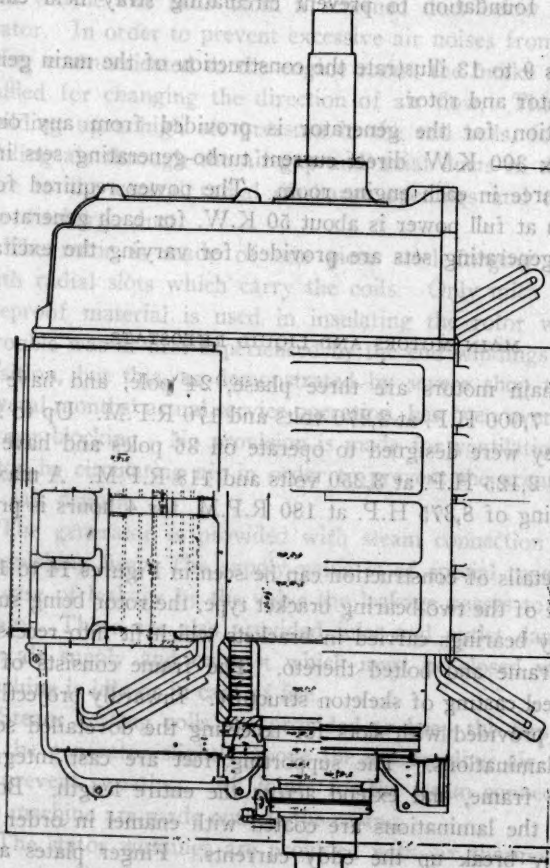


FIG. 9.—LONGITUDINAL SECTION OF MAIN GENERATOR.

There are two independent windings on the stator, one for 24 pole and one for 36 pole. While this makes a more expensive and heavier construction it has the advantage of simple end connections and provides for putting the machine quickly back in service by cutting out one or more coils in

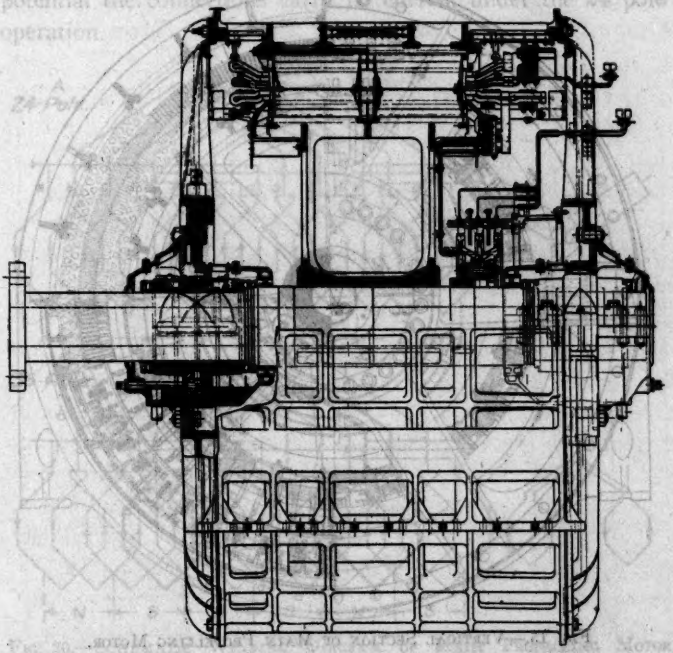


FIG. 14.—LONGITUDINAL SECTION OF MAIN PROPELLING MOTOR.

case of local damage. The coils are form wound and placed in straight open slots, having their ends secured to insulated steel supporting rings which are rigidly supported from the stator frame. The coil for the 24 pole winding consists of four copper straps in parallel. The coil for the 36-pole winding consists of a single copper strap. The straight part of the coil is insulated with mica, only sufficient binding material

being used to hold the mica, and then ironed in a special machine. This produces insulation of the very highest type. Special care is taken by varnishing and baking to prevent damage by salt moisture.

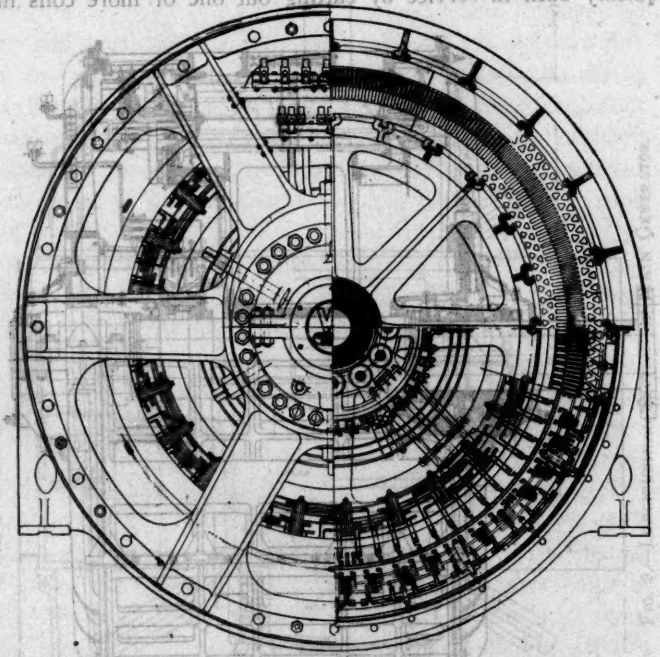


FIG. 13.—VERTICAL SECTION OF MAIN PROPELLING MOTOR.

The rotor is wound with one three-phase winding, permanently connected to three collector rings. By special end connections this winding is adapted for either 24 or 36 pole operation. Figure 20 shows a schematic winding diagram illustrating how this is accomplished. Referring to the drawing "A" shows the condition when operating on 24 pole and "B" for 36 pole. It will be noted at "A" that the coils are grouped and connected to form two parallel circuits for each phase.

The phases are connected in star and by connection to slip rings are brought out to electrodes of a liquid rheostat for varying the external resistance. There being two parallel circuits certain equal potential points are formed which are joined by special connectors. These points being of the same potential the connections carry no current under the 24 pole operation.

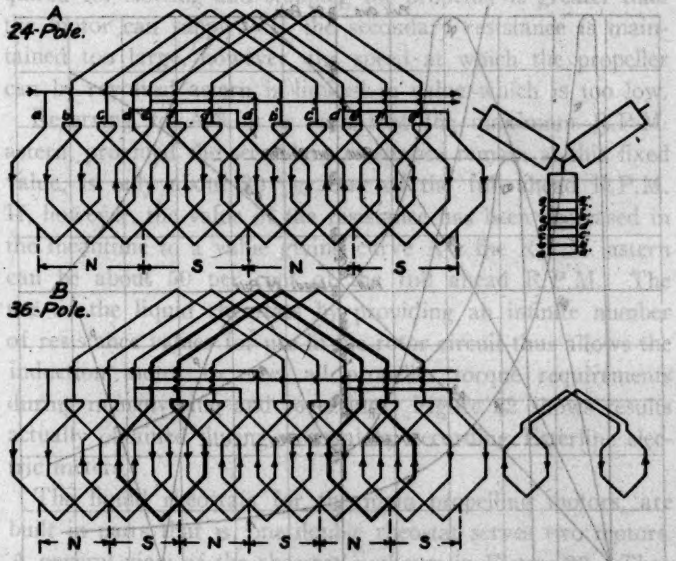



FIG. 20.—SCHEMATIC WINDING DIAGRAM OF MAIN PROPELLING MOTOR ROTOR.

When operating on 36 poles the conditions of the rotor windings are as shown at "B" which also indicates the direction of the e.m.f.'s of the several coils. The special end connectors now serve as short circuits, connecting pairs of coils in series with their e.m.f.'s added. All these coils of the rotor now being short circuited in pairs the motor will have characteristics similar to those of a squirrel cage induction motor.

The propeller torque characteristics make it necessary that special consideration be given to the motor design to meet these requirements. The torque of an induction motor at a given voltage can be varied by varying the resistance in the rotor circuit. To understand the propeller torque requirements and the torque characteristics of the induction motor refer to Figure 21. The propeller torque curve P is shown when re-

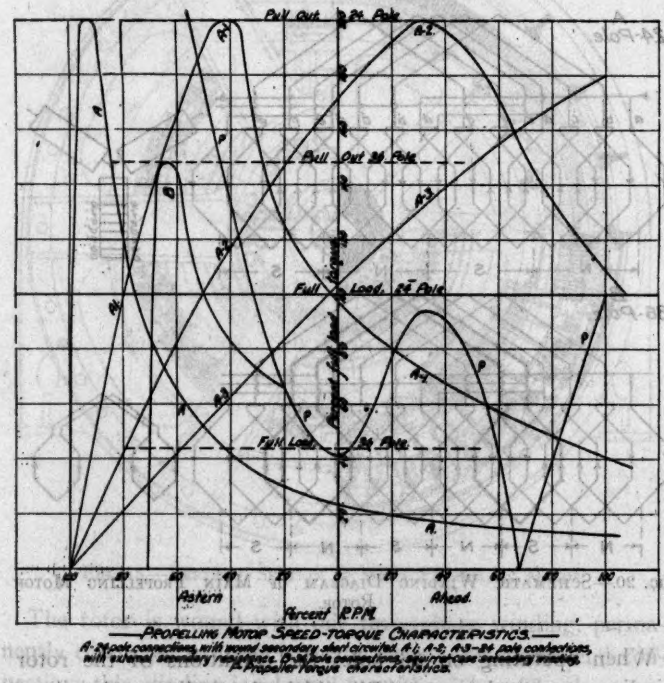



FIG. 21.—PROPELLER AND MOTOR SPEED-TORQUE CHARACTERISTICS.

versing the propeller with the ship going ahead at full speed. The assumption is made that throughout the act of reversal the ship maintains full speed through the water. A; A-1; A-2; A-3 show the motor speed—torque curves for different

values of secondary resistances. A is with minimum secondary resistance, that is, rotor windings short circuited at the slip rings. A-1, A-2, A-3 are with increasing values of secondary resistances, inserted into the rotor winding by the use of external resistance in the form of a liquid rheostat. By a study of the Figure it can be seen that if the value of the secondary resistance is too low, curves A and A-1, torque required for slowing and stopping the propeller is greater than the motor can supply. If the secondary resistance is maintained too large, however, the speed at which the propeller can be revolved astern is limited in value which is too low.

Referring to A-3, it is seen that the maximum R.P.M. astern, provided the secondary resistance remain at this fixed value, is only about 20 per cent of the full ahead R.P.M. If, however, the value of the resistance has been decreased in the meantime to a value giving curve A-1 the R.P.M. astern can be about 50 per cent of the full ahead R.P.M. The use of the liquid rheostats by providing an infinite number of resistance values for use in the rotor circuit thus allows the induction motor to meet all propeller torque requirements during maneuvering and reversing. Figure 22 shows results actually obtained during reversal by recording Esterline electric meters.

The liquid rheostats for the main propelling motors, are built in pairs that is, one double rheostat serves two motors. A general view of the rheostat is shown in Figure 23. They consist of a lower electrolyte storage tank, and two upper electrode tanks. The lower tank is connected with the electrode tanks through centrifugal pumps, and there is an overflow from the upper to the lower tanks so that the electrolyte can circulate. The centrifugal pumps pump the electrolyte from the storage tank to the electrode tanks at a pre-determined rate which can be adjusted by means of a valve in the pump suction. This fixes the rate of acceleration. For normal operation it takes about 17 seconds from minimum to

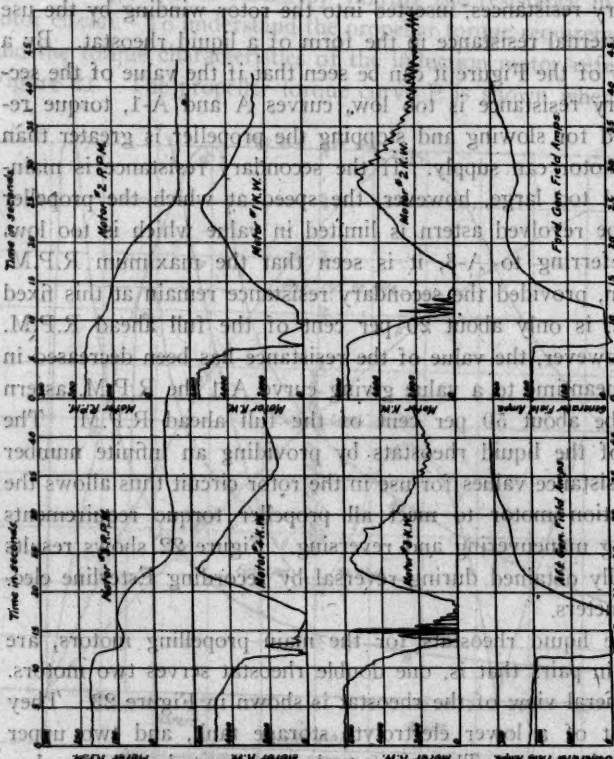



FIG. 22.—REVERSING CURVE PLOTTED FROM RESULTS OF ESTERLINE ELECTRIC RECORDING METERS.

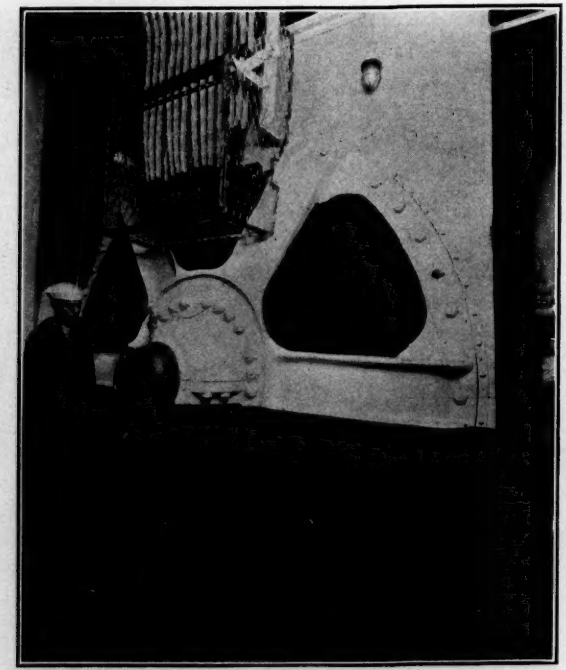


FIG. 19.—MAIN PROPELLING MOTOR AS INSTALLED IN THE SHIP.

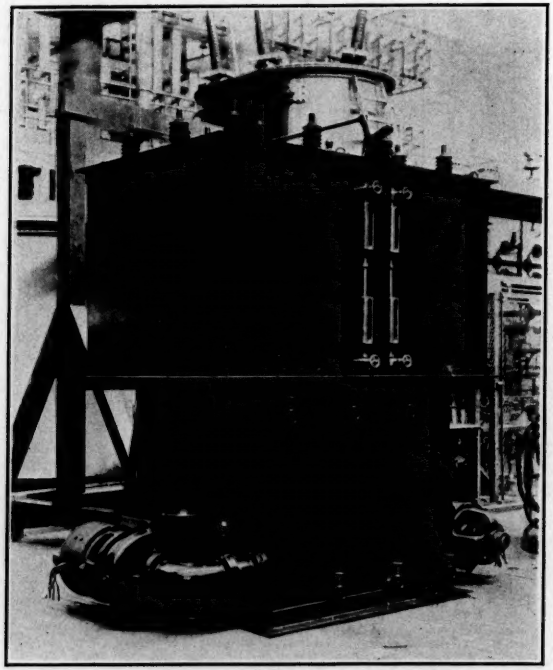


FIG. 23.—DOUBLE LIQUID RHEOSTAT.

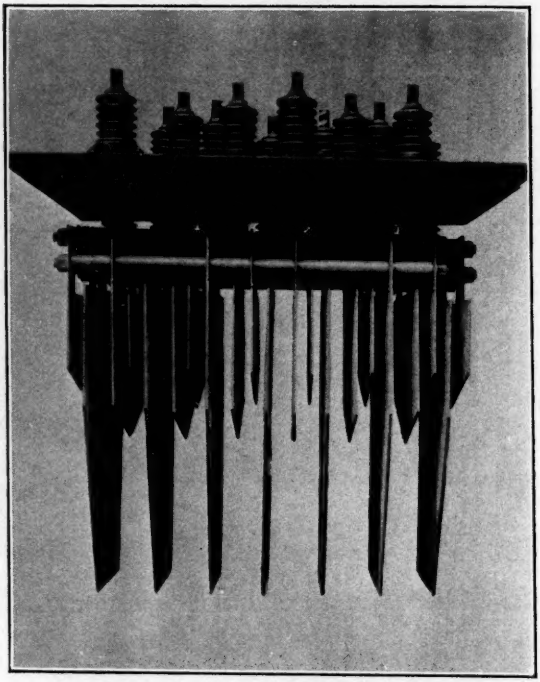


FIG. 24.—ELECTRODE UNIT OF LIQUID RHEOSTAT.

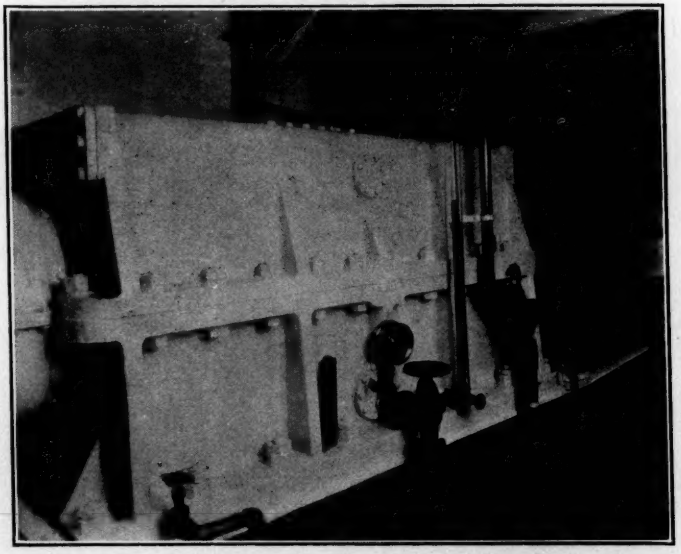


FIG. 25.—KINGSBURY THRUST BEARING HOUSING.

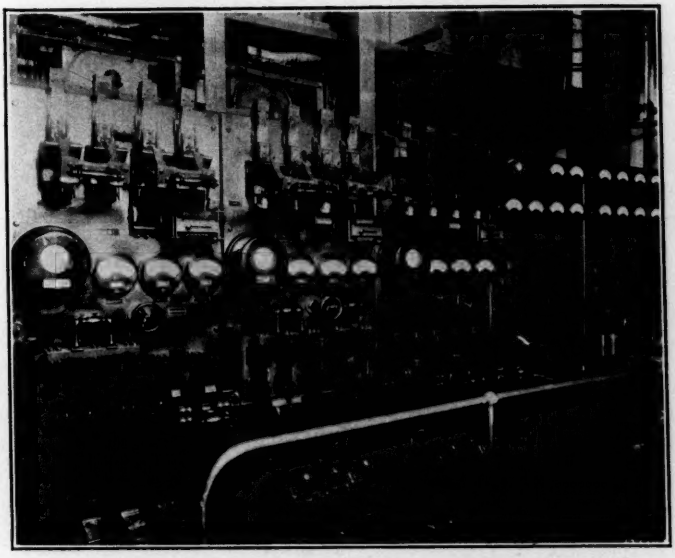


FIG. 27.—WIRING DIAGRAM OF ENGINE ROOM AUXILIARY SWITCHBOARD.

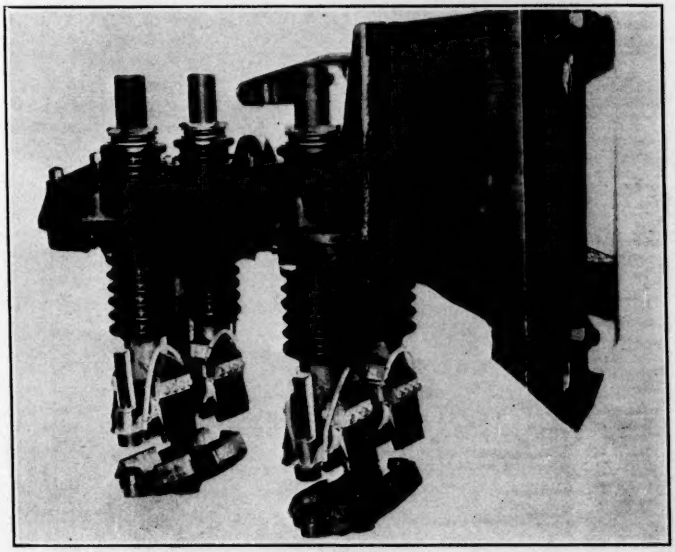


FIG. 29.—VIEW OF TWO-POLE OIL BREAKER.

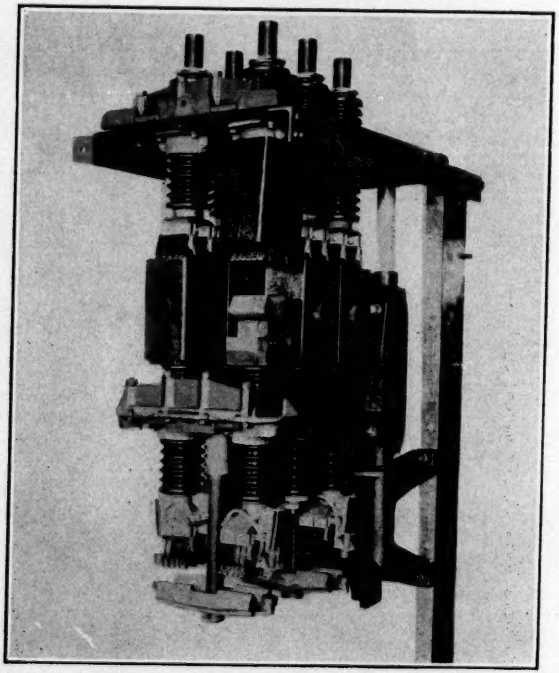


FIG. 30.—VIEW OF THREE-POLE OIL BREAKER WITH DISCONNECTS.

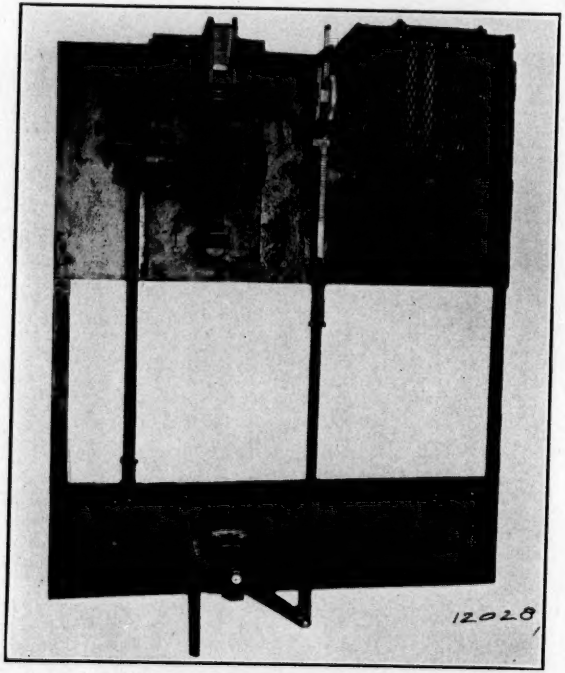



FIG. 31.—MAIN GENERATOR FIELD SWITCH AND BOOSTER RHEOSTAT

maximum level and about 15 seconds for the reverse operation. The electrode tanks have a valve at the minimum liquid level which, when open, allows the electrolyte to circulate through the tank with a part of the longest electrodes immersed. This is the starting position of the rheostat. When it is desired to accelerate this valve is closed through the operating handle and the level of the electrolyte rises until the overflow is reached. At this point the electrolyte will continue to circulate and there is minimum resistance in the motor circuit. Provision is made by means of a handle at the top of the rheostat to connect two electrode chambers together. This can be done either for equalizing the level of the liquid, or in case one of the pumps is out of commission, one pump can fill both chambers at a slower rate.

The lower tank is provided with cooling coils which may be used if it is found necessary due to operating conditions, although ordinarily they will not be required. The electrodes are iron plates supported by special moulded insulators and can be removed as a unit. The construction of the electrode can be seen in Figure 24. A vent pipe is provided to allow the escape of any vapor.

The liquid used in the rheostats is a 1½ per cent solution of Sodium Carbonate ($Na_2Co_3$), known commercially as Soda Ash. The resistance between phases is about .08 ohms, with maximum liquid level and .35 ohms with minimum level.

Thermometers are provided for indicating the temperature of the electrolyte and gauge glasses for indicating its level.

The following data describes the pump used for circulating the liquid:

Pump, type. Gould Manufacturer Company's horizontal, centrifugal.
speed, R.P.M. ..... 750
capacity, gallons per minute ..... 750
head, feet ..... 7
Motor, horsepower ..... 5
Control, contactor type with overload and no-voltage protection.

FIG. 30.—VIEW OF THREE-POLE OIL BREAKER WITH DISCONNECTS. FIG. 31.—MAIN GENERATOR FIELD SWITCH AND BOOSTER RHEOSTAT

In arranging the motor to perform all maneuvering on the 24 pole connection, the turbine speed does not have to be changed for shifting to 36 pole operation. This is due to the fact that when power is removed the propeller R.P.M. drops immediately to about 70 per cent. This insures a quick shift which is sufficiently rapid to allow it to be done in fleet formation without losing position. Severe strain is also prevented on the windings during the switching operation.

For gaining access to all parts of the motor for examination or repair, the foundations are extended to the forward bulkhead. The stator can be slid forward on these extended foundations a sufficient distance to completely uncover the rotor. The motor feet are grooved to fit a key laid and bolted to the foundations, which keeps the stator moving parallel to its axis. This insures the windings against damage even with the small air gap of .120 inch. In order to disassemble it is necessary to (1) unbolt the after bearing bracket; (2) remove the forward upper bearing housing and bearing; (3) fit an extension shaft, one end inserted into the motor shaft and the other end resting on a saddle foundation at the forward bulkhead; (4) unbolt and lift clear the blower motors, foundations and ventilating ducts; (5) disconnect and clear the cable connections, (6) remove incidental piping and gear such as oil piping, counter gear, floor gratings and supports. After practice a motor has been disassembled, examined, reassembled, and put back in service in less than twenty-four hours.

The stator windings are provided with six thermo couples, three for each winding, located in the slots between top and bottom coils for the purpose of measuring the hot spot temperature. The leads from the thermo couples are lead to the control room so that temperature readings from all main units can be read at one point which is under the direct supervision of the watch officer.

The shaft is of nickel steel, with the coupling forged integral. The shaft is hollow, having a nine-inch hole in the center throughout its length.

The three collector rings are of brass and are assembled on a cast steel hub which is pressed on and keyed to the shaft. Special care is taken in the assembly of the rings to provide highest type of insulation. The brush rigging is assembled on a suitable casting and bolted to the forward bracket. A fitted sheet steel pan is installed beneath the brush riggings so that any loose parts or broken brushes will be prevented from dropping into the windings.

Ventilation is provided by two 750 to 1,200 R.P.M., 10,000 cubic feet per minute, variable speed, motor-driven exhaust fans. They are mounted on top of a sheet steel casing which incloses the motor frame and serves to confine and direct the cooling air. In addition the motor rotor is provided with fans so that in case of failure of the separate motor-driven fans operation could be continued for a short period. The cooling air enters the motor room through an armored natural ventilating duct which leads down from the quarter deck. At the base of this duct, a false watertight deck is installed over the cables and buss structure to prevent damage from any water which might enter. The exhaust blowers draw the cooling air from the room through screened openings in the motor brackets, over the coil ends, through the air gap and axial ducts in the stator and rotor cores formed by punched holes in the laminations, and thence through a radial duct located centrally in the cores to the chamber formed by the sheet metal casing around the frame. Here the air enters the exhaust fan which forces it up through an armored duct where it is discharged on the quarter deck. Provision is made against damage by water entering the motor from the discharge duct by a large trap at its base which has a drain to the bilge. All discharge and supply ducts on the weather deck are carried well above the deck and fitted with special channel deflecting

shields around the trunk and with a special type hood over the trunk. In very cold weather the motor room is heated by passing a part of the heated air discharge from the blower back to the room. An adjustable shutter in the discharge duct is provided for this purpose. The intake duct to the blower is also provided with damper so if necessary only one blower can be operated.

The bearings are of the spherical seat, self-aligning type and are made of cast steel, babbitt lined. They are split and designed so that the top half can be removed for examination of the journal without disturbing the bottom half. The bearings are carried in cast steel housings, also split through the horizontal diameter, and these housings are bolted to the cast steel brackets. By use of jack screws the position of the housings can be adjusted radially in the brackets for the purpose of centering the rotor and thus adjusting the air gap.

Usually lubrication is supplied from the after engine room lubrication system. In case of failure of this system the bearings are provided with oil rings and designed to operate indefinitely at full power without overheating. The oil reservoir is provided with a thermometer and an illuminated sight flow indicator in the oil drain. Special oil guards are provided to prevent oil from passing out of the bearing housing into the motor.

Direct current heaters are installed in the motor frame in order to prevent condensation in the motor when it is lying idle.

On March 12, 1921, number 4 main motor was damaged to such an extent by a short circuit that it was necessary to completely rewind both rotor and stator. The short also, due to the intense heat developed, fused three of the slots in the stator so that about ten per cent of the iron laminations had to be renewed. The fused portion of the slots extended from the after end to the central radial air duct. After giving all phases of the question of rewinding careful consideration it

was decided to do the work on board ship and away from a Navy Yard. The ship was at Guantanamo, Cuba, at the time of the casualty so proceeded to Hampton Roads, Va., where the material was assembled. The work was done under the direct supervision of the contractor's service engineer. The direct labor was performed by eight winders and one iron worker from the Westinghouse works, assisted by picked men from the engineer force of the ship.

Complete rewinding would not have been necessary except that no provision had been made for extinguishing a fire in the windings of the motor so that salt water had to finally be used for that purpose. All motors, both in service and new construction, will be fitted with a special sprinkling system connecting with the boiler water feed mains for extinguishing a fire resulting from a short circuit. In this way coils not damaged mechanically will be saved.

Although the exact cause of the accident could not be definitely determined it has been practically agreed that it was caused by some foreign substance getting between the moving and stationary coils, thus injuring the insulation. The wire mesh screening fitted over the openings in the brackets for the entrance of the cooling air was of very coarse mesh, about 1 inch opening. This was replaced on all the motors with heavy screening, having about ⅛-inch opening.

During the rewinding of number 4 motor the other three motors were completely disassembled, examined carefully, sprayed with varnish, given a high potential test, and reassembled by the ship's force. This was in order to provide every precaution against the recurrence of the casualty to number 4 motor.

The question of baking the varnish properly after spraying the windings was quite a problem to accomplish on board ship. This baking had to be done at about 100 degrees Centigrade and six coats of this insulating varnish had to be sprayed and baked on. This was effectively accomplished by using twenty-

five three-thousand-watt electric heaters, supplied for heating various parts of the ship, and using the entire motor room for an oven.

During the progress of the work the ship proceeded to the Naval Proving Grounds at the mouth of the Potomac River and carried out the test firing of her entire battery. After this she proceeded to Boston, Mass., for docking, previous to standardization and machinery trials, and thence to Rockland, Me., arriving there on May 12, 1921. It was thus only two months from the breakdown in Cuba until ready for final trials at Rockland.

With the repairs in progress it was possible, as demonstrated by actual runs, for the ship to steam in the open sea up to speeds of about 15 knots.

## SHAFTING AND BEARINGS.

There are four lines of shafting, each driven by one of the main propelling motors. Each line of shafting is composed of the following number of separate sections, all rigidly coupled together.

| | Inboard Shafts. | Outbroad Shafts. |
|---|---|---|
| Thrust shaft | 1 | 1 |
| Line shaft | 2 | 2 |
| Stern tube shaft | 1 | 1 |
| Propeller shaft | 1 | 1 |

*Thrust Shafts*—The thrust shafts are fitted to thrust bearings of the Kingsbury type, with steady bearing near the forward coupling of the shaft and also a steady bearing at each end of the thrust bearing. The coupling discs and thrust collar are forged integral with the shaft.

*Line Shafts*—These are hollow forged, with disc couplings at each end forged integral with the shaft. The after section of the outboard line shafting and both sections of the inboard line shafting are carried in two steady bearings, one

near each end of the shaft. The forward section of the outboard line shafting is carried in a single steady bearing near its center.

*Stern Tube Shafts*—They are hollow forged, secured to the line shafts by special disc couplings in order to provide means for removal through the stern tube bearing. They have composition casings shrunk on where they pass through the stern tubes and stern-tube bearings. Special sleeve couplings are used for securing to the propeller shafts. Two lignum vitæ bearings are provided for each stern tube shaft. A standard stuffing box gland is provided where the shaft passes through the forward bulkhead, for inboard shaft at frame 124 and outboard shaft at frame 115. Fairwaters are fitted where the shaft leaves the after stern tube bearing.

*Propeller Shafts*—They are hollow forged, taper turned at after end to suit the propeller hub, and have composition casing shrunk on where they pass through the strut bearing. Each shaft is carried by a single lignum vitæ-lined strut bearing. Fairwaters are fitted at each end of the strut bearing.

*Inboard Coupling*—The stern tube shaft is upset and has fitted over this portion a sleeve secured by four keys. Back of the sleeve is a collar made in halves and fitted against the shoulder formed by the upsetting of the end of the shaft. These collars are secured to the sleeve and line shaft disc by through bolts.

*Outboard Coupling*—These couplings are of the solid sleeve type, taper bored for the reception of the shafts, and secured by two feather keys and one cross key.

The line shafts and thrust shafts where they pass through watertight bulkheads have a special type sleeve stuffing box. The opening in the bulkhead is made oversize and a sleeve, having a standand type stuffing box gland fitted in its end, is carried to the shaft foundations where it is supported. This allows a deflection of the bulkhead without binding the shaft.

## SHAFT DATA.

| | |
|---|---|
| Thrust shafts, length, inboard, feet and inches | 24– 0 |
| length, outboard, feet and inches | 21– 0 |
| diameter, inches | 15 1/4 |
| diameter, at bearings, inches | 15 1/2 |
| diameter, axial hole, inches | 7 5/8 |
| thrust collars, number | 1 |
| thrust collar, thickness, inches | 6 1/2 |
| outside diameter, inches | 29 3/8 |
| inside diameter, inches | 15.740 |
| Line shafts, forward, inboard, length, feet and inches | 28– 0 |
| after, inboard, length, feet and inches | 25– 0 |
| forward, outboard, length, feet and inches | 21– 0 |
| after, outboard, length, feet and inches | 21– 0 |
| diameter, inches | 15 1/4 |
| at bearings, inches | 15 1/2 |
| axial hole, inches | 7 5/8 |
| Stern tube shafts, length, inboard, feet and inches | 44– 2.548 |
| outboard, feet and inches | 43– 2.522 |
| diameter, inches | 16 7/8 |
| at coupling, inches | 18 3/8 |
| at bearings, inches | 18 5/8 |
| axial hole, inches | 8 3/8 |
| axial hole, at outboard coupling sleeve, inches | 4 |
| sleeves, thickness at bearings, inch | 7/8 |
| taper in outboard coupling, inches per foot | 1 |
| Propeller shaft, length, feet and inches | 31– 1 11/16 |
| diameter, inches | 16 7/8 |
| diameter, axial hole, inches | 8 3/8 |
| diameter, axial hole at propeller hub and outboard coupling, inches | 4 |
| taper in outboard coupling, inches per foot | 1 |
| taper in propeller hub, inches per foot | 1 |
| Coupling discs, diameter, inches | 30 |
| thickness, inches | 3 1/2 |
| number of bolts | 8 |
| Inboard coupling, diameter of sleeve, outside inches | 30 |
| inside, inches | 18 3/8 |
| length of sleeve, inches | 13 |
| thickness of collar, inches | 3 1/2 |
| Coupling bolts, number each coupling | 8 |
| diameter (taper) at face of coupling | 3 1/4 |
| Outboard, coupling, length of sleeve, inches | 63 1/4 |
| diameter, outside, inches | 22 1/2 |

BEARING DATA.

| | |
|---|---|
| Steady bearings, thrust shafts, number | 3 |
| diameter, inches | 15 1/2 |
| length, one forward, inches | 26 |
| length, two within thrust bearing housing, inches | 16 |
| Line shaft bearing, diameter, inches | 15 1/2 |
| length, inches | 26 |
| Stern tube bearing, forward, diameter, inches | 18 5/8 |
| length, inches | 48 |
| after, diameter, inches | 18 5/8 |
| length, inches | 60 |
| Strut bearings, diameter, inches | 18 5/8 |
| length, inches | 71 1/2 |

*Thrust Bearing*—This bearing is a standard 29 inches Kingsbury type and contained in an oil tight housing with two steady bearings. A general view of the bearing housing, as installed, is shown in Figure 25. The collar forged integral with the shaft has bolted on each side face plates which bears against the shoes. These face plates are made in halves so they can be readily renewed. The face of the bearing shoes are babbitted, but a spare set of bronze shoes are carried. The shoes are carried on leveling plates which insure equal division of the load between the shoes. The housing is supplied with oil from the after forced lubrication system and has an overflow in the top which causes it to be continually full of oil. A water circulating cooling coil is fitted in the top of the housing to assist in carrying off the heat. The bearing was designed to operate at full power with failure of both lubricating oil and water supply. It has been recently demonstrated on full power trial of the U. S. S. *Maryland* by actual test that this can be done.

PROPELLERS.

There are four three-bladed propellers, cast with blades integral with the hub. The starboard propellers are right-hand and the port propellers left-hand, all propellers turn outboard

when driving the ship ahead. They are of manganese bronze and the hubs are taper bored to fit the shafts, to which they are secured by a key and locked nut. The ahead face of the blades are machined true to pitch and the propeller polished all over including the hubs. They are given a static balance on knife edges and sufficient metal removed from the backing face of the blades to obtain balance.

PROPELLER DATA.

| | | |
|---|---|---|
| Diameter, feet and inches | 13 | – 6 |
| hub, feet and inches | 2 | – 9 |
| Pitch, feet and inches | 14 | – 3 |
| Ratio of diameter to pitch | | .9474 |
| Area, projected, square feet | 52.1 | |
| helicoidal, square feet | 61.98 | |
| Ratio, projected to disc area | | 3.64 |
| helicoidal, to disc area | | .4333 |
| Height of lower tip of blade above keel, inboard, inches | | 9 |
| outboard, inches | | 17.17 |
| Immersion of upper tip of blade, inboard, feet and inches | 16 | – 0 |
| outboard, feet and inches | 15 | – 3.83 |

CONTROL EQUIPMENT.

The control equipment consists generally of the following:

(1) One direct current engine room auxiliary switchboard in each engine room.

(2) Alternating current switching structure in control room which has mounted within it all motor and generator oil circuit breakers and contains the buss and cable connections to and from these switches.

(3) One contactor with field discharge resistance and booster field rheostat for each main generator, located in the control room.

(4) Group of turbine, generator and motor operating levers, located in the control room.

(5) An instrument board, mounted on the alternating current switching structure, having on it all electric, steam, water and oil gauges necessary for furnishing information to the operator directing the operation of the propelling machinery.

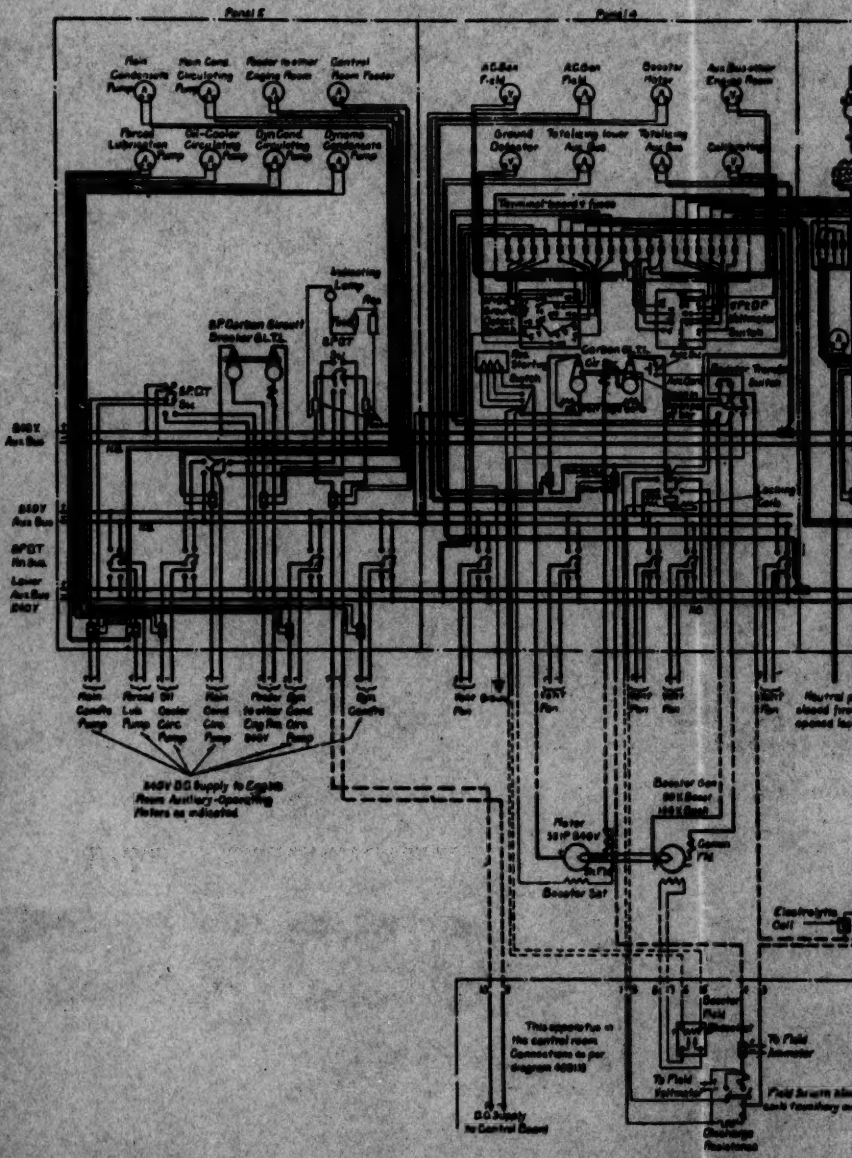
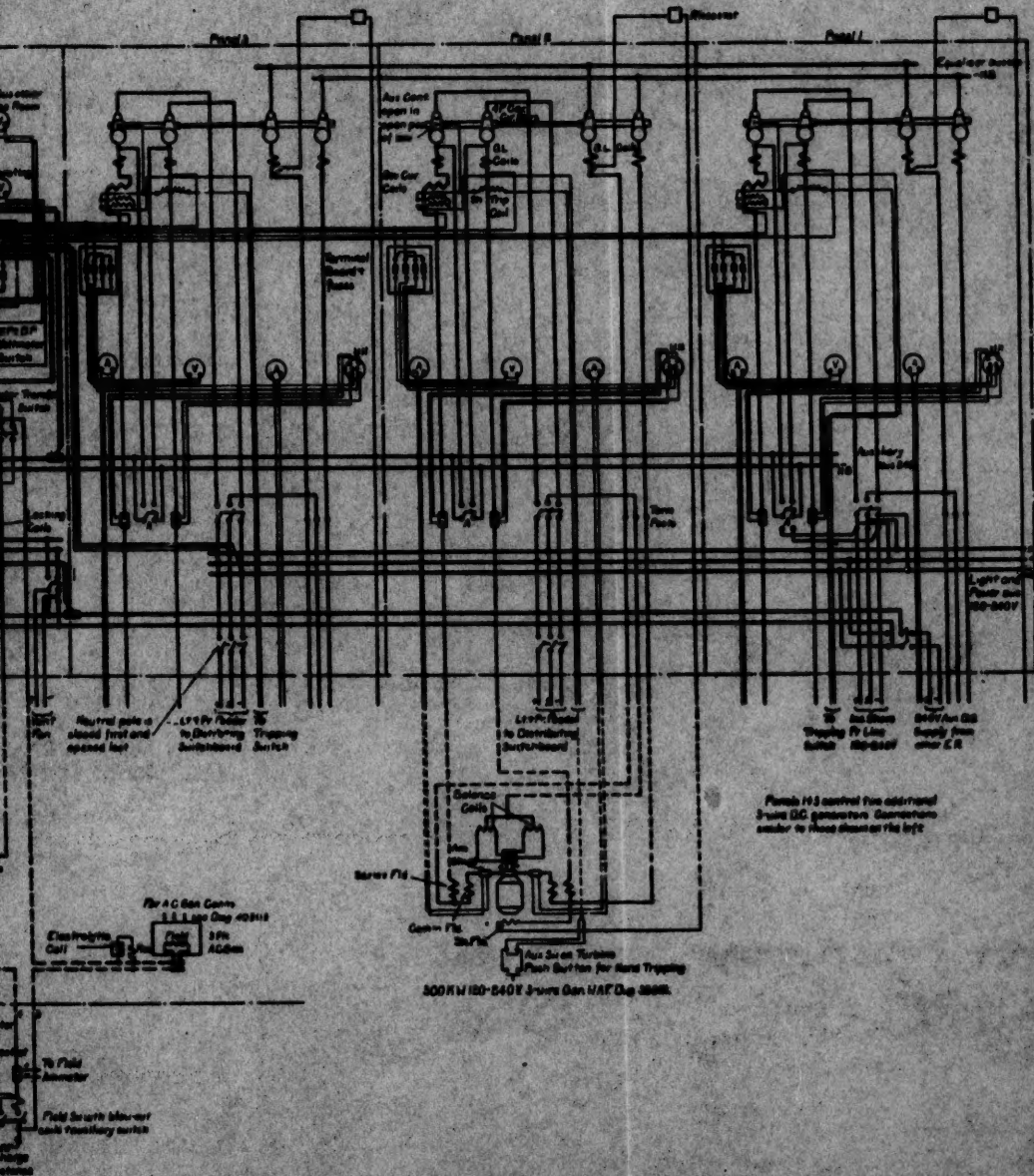


FIG. 26.—FRONT VIEW OF EN


Panel A
Panel B
Panel J
Reverse Current Trip Coils
Trip Coil
Auxiliary Bus Bars
Neutral pole is placed first and opposed last
To Distributing Switchboard
To Distributing Switchboard
Balancer Coils
Series Fld
Comm Fld
Shunt Fld
Aux Steam Turbine Push Button for Hand Tripping
To Field Rheostat
Panels 1-3 control five additional 3-wire D.C. generators. Connections similar to those shown at the left

VIEW OF ENGINE ROOM AUXILIARY SWITCHBOARD.

(6) Liquid rheostats, with their electrolyte pumps and controllers, for maneuvering the main motors.

(7) Communication equipment consisting of telephones, telegraphs, voice tubes, telautograph, etc., for receiving and transmitting orders.

(8) Fireroom central control board located in control room, for controlling the boiler operation.

Each of the direct current auxiliary switchboards controls and distributes the current from the three 300 K.W. 120/240 volt, three wire direct current turbo-generator sets. Figure 26 shows the diagram of connections of this switchboard and Figure 27 a photograph of the after engine room board. Each switchboard has a set of three wire light and power bus-bars and two sets of two-wire main drive auxiliary bus-bars. The light and power feeders to the ships distribution board are supplied from the three wire bus-bars.

By the use of the three pole, double-throw switch, shown on the diagram, power can be supplied to the ship from an outside source. Permanent cable leads are made from this switch by two connections, one on each side of the quarter-deck for connecting to shore power supply.

As will be noted from the diagram any one of the three generators can be used for excitation and main drive auxiliaries. The auxiliary busses are connected direct to the generator without interposing fuses or circuit breakers of any kind. This is done on account of the necessity of absolute reliability for supply of power for excitation and main drive auxiliaries. Switches marked "A" are so interlocked that only one can be thrown in at any one time. This is done on account of the danger to the generator when paralleling without protective devices in circuit. The interlock is arranged, however, so it can be made inoperative temporarily for paralleling when it is desired to shift another machine to this buss. The interlock is self-restoring. All circuit breakers are equipped

with time limit device and shockproof feature. They also have a shunt trip coil which is operated by the turbine emergency overspeed trip.

The upper auxiliary buss is the primary system. The lower buss is kept energized from the light and power generator in the same engine room in case there is no generator operating in the other engine room. In case a generator is in operation in the other engine room then the lower buss is energized from the upper auxiliary buss of that room. This is done in order to have available at all times two sources of power.

The board is supplied with the necessary ammeters and voltmeters as shown. Standard two-pole double-throw knife switches are used to supply the main drive auxiliaries with power from either buss. The larger switches, such as the main circulating pump, have special blades in order to reduce the time of throw-over. This is done so that their source of power can be changed without interrupting the motor operation.

The control for the booster generator set for supplying excitation for the main generators was given special consideration. The requirement was made that excitation could be switched directly to one of the 300 K.W. generators without interrupting the operation of the main generator. This is done by providing a two-pole double throw transfer switch, which does not open the circuit during transfer. This switch is provided with an auxiliary pole which opens the circuit to the booster field rheostat, so that the booster generator cannot be short circuited with its field energized in case the switch is thrown before tripping the motor circuit breaker. A signal is fitted from the control room to the board for notifying the operator to make this shift. After the shift is made all auxiliaries can be transferred to the reserve buss and the excitation varied by controlling the exciter voltage by use of its rheostat. The booster motor is started by a single pole, four point hand-operated switch. It is protected by two pole circuit break-

ers which upon opening also opens the booster generator field. This is done to prevent the booster set from running away in case the breakers open.

A double-throw switch is provided for connecting the main generator field circuit and the booster set circuit to either auxiliary buss. The switch has a solenoid operated latch on each throw, which prevents opening the switch unless the field lever in the control room is in the "off" position.

Figure 28 shows a schematic wiring diagram of the alternating current and direct current circuits.

The alternating current switching structure has its lower part divided into two rows of eleven flameproof compartments, each of which contains an oil circuit breaker.

These oil breakers consist of the following:

4—1,600 ampère, two pole breaker, with three pole disconnects for generators.

8—1,600 ampère, two pole breaker, with three pole disconnects for motor ahead and astern.

2—800 ampère, three pole breaker, for motor ties.

4—1,600 ampère, three pole circuit breaker for 24 pole connection.

4—800 ampère, three pole circuit breakers for 36 pole connection.

In two structures separate from the main structure, each having two flameproof compartments, are 4—2,000 ampère, two pole breakers for short circuiting the motor secondaries.

The type of breakers and disconnects used can be seen in Figures 29 and 30. The insulators are of molded bakelite. Insulators throughout all of the equipment are of this or similar material, having nonfragile properties. An interlock is provided between the breaker and disconnects so that the disconnect cannot be opened unless the breaker is open and the breaker cannot be closed unless the disconnects are closed. Provision is made for disconnecting the motor oil breakers from

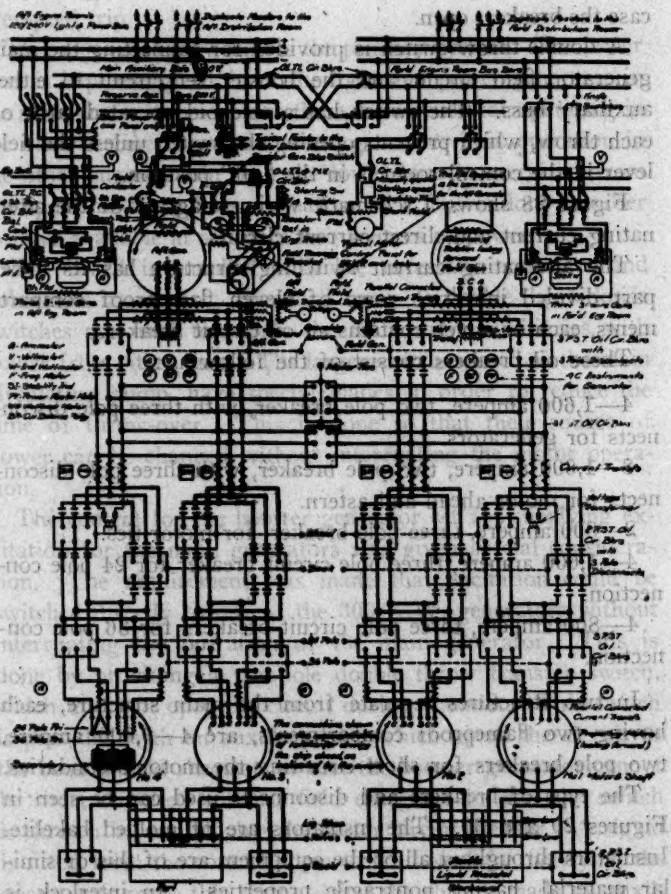


FIG. 28.—COMBINED ALTERNATING CURRENT AND DIRECT CURRENT WIRING DIAGRAM.

their operating lever so any motor can be disconnected and operation continued on the remaining motors as desired.

The top of the structure supports the necessary buss work and the incoming and outgoing cable terminals. The cables terminate in special triplex pot-heads which have a wiped lead joint with the cable sheath. The pot-heads are filled with insulating gum and the single cable outlets specially taped and passed through bakelite insulators in order to render them moisture proof. All bus structure is screened in order to prevent possible injury to personnel.

The field contactor and booster generator rheostat are manually operated by one lever. The contactor is closed on maximum field voltage and opened on minimum voltage. The rheostat has a single resistance connected across the 240-volt auxiliary bus and various resistance taps are brought out and connected to two face plates. The voltage across the booster generator field is practically zero when the arms on the face plate points directly downwards. When the arm is moved in one direction, voltage of one polarity is gradually increased across the booster field and when moved in the opposite direction is of opposite polarity. The booster generator thus bucks or boosts the exciter voltage and excitation voltage can be adjusted to any desired value by proper setting of the field control lever. A view of the field switch with rheostat is shown in Figure 31.

The main generators, main motors, alternating current cables and switching equipment is protected from serious damage due to ground or short circuit by current unbalance protection and by a ground detector device.

An unbalance on any motor circuit will, through the current unbalance protection, cause the tripping of an automatic field switch of the generator supplying it with power. This is accomplished by current transformers installed on each phase of each motor which, in case of unbalance, causes the operation of a three-phase balance relay. When this relay

operates it closes a direct current circuit through a no-overload shunt trip which opens the generator field circuit, thus removing power from the line. Arrangements are made by the use of auxiliary switches on the generator circuit breaker mechanism for tripping only the field breaker of the generator developing the fault. Provision is made by test switches installed on the instrument board for testing the operation of the entire system at any time. The automatic field switch is installed in series with the operating field switch and is of the hand closing and shock proof type. In case of opening, the generator field is discharged through the same set of field discharge resistance as used for the operating field switch. An indicator on the auxiliary relay will indicate which motor circuit is in trouble.

Provision is made through the use of a knife switch for opening the shunt trip circuit for the automatic field switch to make the automatic protection ineffective whenever it is desired.

A ground on any motor primary circuit or on either generator will, through the operation of the ground detector system, cause an alarm bell to ring. This is accomplished by three voltage transformers, with one primary connection to each of the main generator phases and with the other primary leg individually grounded. The transformer secondaries are connected in open delta. From this it is apparent that as long as the generator phases are clear of ground the voltage across the open delta is zero, but in case of ground on any phase the voltage across the open delta is of a value depending on the resistance to ground. In case of voltage across the open delta an alternating current relay is operated, which, by closing a direct current circuit, rings an alarm bell through the action of a direct current relay. Provision is made so that by proper switching all parts of the system can be tested for operation.

By reference to Figure 28 showing the combined alternating current and direct current diagram, all circuits of both systems can be traced.

In order to limit the induced voltage of the generator field upon breaking its circuit a field discharge resistance is connected across the lower set of contacts. These contacts close, thus cutting the field discharge resistance in circuit, just before the field contacts open that circuit. A duplicate field discharge resistance in series with an electrolytic cell is connected directly across the main generator field slip rings for additional protection in case the field switch fails to function properly.

The electrolytic cell consists of aluminum plates in a special electrolyte and acts similarly to a nonreturn valve. When the exciting circuit is opened by means of the field switch, the cell breaks down, due to reversed polarity across its terminals and allows the field energy to be absorbed by the resistor. The reversed polarity is due to the fact that the current induced upon the collapse of the field flows in the same direction as the exciting current which was interrupted.

The group of turbine, generator and motor operating levers located in the control room consists of the following:

(1) One special control stand for each main turbine, with power limit operating controller.

(2) Turbine emergency trip, one for each turbine.

(3) One field control lever for each main generator.

(4) One motor tie breaker lever.

(5) Two generator breaker levers, one for each generator.

(6) Four motor direction levers, two (ahead and astern) for the two starboard motors, and two (ahead and astern) for the two port motors.

(7) Four pole changing levers, two (24 and 36 pole) for the two starboard motors and two (24 and 36 pole) for the two port motors.

(8) Two liquid rheostat valve levers, one for starboard motors and one for port motors.

(9) Two motor secondary short circuiting levers, one for starboard motors and one for port motors.

All generator and motor levers operate their breakers by direct mechanical connection. They are so interlocked as to make it impossible for the operator to move a lever which would damage the apparatus. All interlocks are mechanical and made very rugged. Levers are painted a distinctive color to make it easier for the watch officer to observe the set-up. The ahead and astern levers are colored white and red, respectively, to correspond to the engine order telegraph dials and equipped with a buzzer signal to give an alarm in case the lever is thrown to a setting which does not correspond to the order on the telegraph. The arrangement of the operating levers and interlocks is shown in Figure 32.

The motor tie connects the two sets of motor bus-bars in the control room so that all four motors can be operated from one generator.

Each generator breaker lever operates the two generator breakers which connect that generator bus-bar to the motor bus-bars.

The reversing levers reverse two phases to the motors, thus changing their direction of rotation.

The functions of the turbine speed control stand, with power limit, field control levers, pole changing levers, liquid rheostat valve levers, and motor secondary short circuiting levers are described elsewhere in this article.

Figure 33 shows a general view of the control room.

## OPERATION.

*Condition I.—Stopped and Standing By.*

1. With one generator in operation that generator breaker and motor-tie breakers are closed. With both generators in operation both generator breakers are closed and motor tie breaker open.

2. Either one or both speed control wheels are set for slow speed, depending on number in operation.

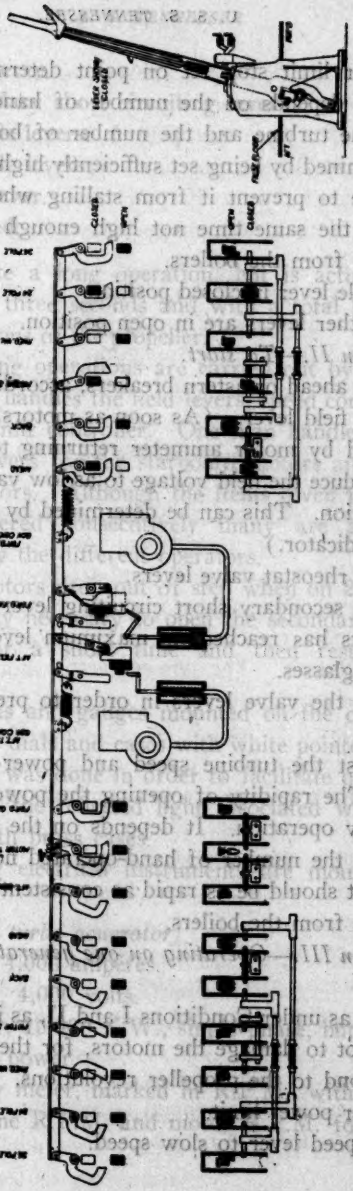


FIG. 32.—OPERATING LEVERS AND INTERLOCKS.

3. Power limit stop set on point determined by practice. This point depends on the number of hand-operated nozzles open on the turbine and the number of boilers in operation. It is determined by being set sufficiently high to admit steam to the turbine to prevent it from stalling when load is thrown on and at the same time not high enough to cause carrying over water from the boilers.

4. 24 pole lever in closed position.

5. All other levers are in open position.

*Condition II.—To start.*

1. Close ahead or astern breakers, according to signal.

2. Close field lever. (As soon as motors are in step, which is indicated by motor ammeter returning to normal reading, begin to reduce the field voltage to as low value as possible for safe operation. This can be determined by observation of the stability indicator.)

3. Close rheostat valve levers.

4. Close secondary short circuiting lever as soon as liquid in rheostats has reached its maximum level as indicated on the gauge glasses.

5. Open the valve levers in order to prepare for the next operation.

6. Adjust the turbine speed and power limit for desired R.P.M. The rapidity of opening the power limit is vital to satisfactory operation. It depends on the number of boilers in use and the number of hand-operated nozzles open on the turbine. It should be as rapid as consistent with not carrying over water from the boilers.

*Condition III.—Operating on one generator on 36 pole connection.*

Proceed as under Conditions I and II, as it is first necessary, in order not to damage the motors, for the speed of the ship to correspond to the propeller revolutions.

1. Lower power limit.

2. Set speed lever to slow speed.

3. Open field lever.
4. Open secondary short circuiting levers.
5. Open 24 pole levers.
6. Close 36 pole levers.
7. Close field lever.
8. Adjust speed.

This reads like a long operation, but is actually accomplished in about three seconds and with a total loss of only about 30 revolutions of the propeller.

All maneuvering operations are carried out by three operators. One man handles the field levers, speed control wheels, and the power limit controller. One man handles the set-up and operating levers for the starboard motors and one those for the port motors. Although the items given under operations are numbered consecutively many are accomplished simultaneously by the different operators.

In case the motors drop out of step when on 24 pole combination it is only necessary to open the secondary short circuiting lever for a short time and then resume normal operation.

All instruments and gauges mounted on the control room board have black dials and cases with white pointers and scale markings. This was done in order to facilitate quick reading and to eliminate the reflected light associated with white or silvered dials with brass cases.

The following electrical instruments are mounted on the board.

*For each main turbo generator:*

One ammeter, 4,000 amperes.

One voltmeter, 4,000 volts.

One wattmeter, 15,000 K.W., single phase, but marked for 3-phase balanced power.

One frequency meter, marked in R.P.M., with three scales to indicate turbine R.P.M. and motor R.P.M. for 24 and 36 pole.

One power factor meter, 3 phase.

One stability indicator.

One ammeter, generator field, 400 ampères.

One voltmeter, generator field, 400-volt.

One position indicator for power limit device

One R.P.M. indicator, magneto operated.

*For each main motor:*

One ammeter, two scales, one 1,600 ampères for 24 pole operation, one 800 ampère for 36 pole operation.

One shaft R.P.M. indicator, 200-0-200 magneto operated.

All instrument transformer secondary leads come to terminal boards above the main panels where they are connected to instrument switches with binding posts for connection of test instruments.

The stability indicator indicates by an arbitrary scale when the field being carried is sufficient for the voltage of the generator to enable it to carry the load safely under the given condition to prevent the motors from dropping out of step. The power factor meter gives the operator a good indication of economical excitation. The power factor should be carried as high as consistent with stable operation. Shaft horsepower can be calculated accurately at any time from the wattmeter reading by allowing for the motor efficiency.

The following gauges are mounted on the board:

*For each main turbine:*

Steam gauge, steam chest, 300 pounds.

Steam gauge, first stage inlet, compound, 300 pounds.

Vacuum gauge.

Governor control valve oil supply, 150 pounds.

Oil pressure to governor, 150 pounds.

General gauges. One for each of the forward and after systems.

Main steam line, 500 pounds.

Auxiliary exhaust, compound 40 pounds.

Main feed water, 500 pounds.

Forced lubrication system to bearings, 15 pounds.

*Other instruments and indicators:*

One shaft averaging counter which gives the revolutions of each of the four shafts, average revolutions of the starboard shafts, average revolutions of the port shafts, average revolutions of all shafts, special "A," "B" counter operated electrically from the bridge for obtaining accurately the average revolutions for any distance traveled by the ship, and a special averaging counter for use in connection with a stop watch for obtaining the R.P.M. There is also a movable pointer on a dial which indicate the relative speed of the port and starboard shafts. The counter is connected mechanically to and operated by the propeller shafts.

One potentiometer for reading the hot spot temperature of the main generators and motors. The instrument reads up to 200 degrees Centigrade, and is connected to the thermocouples installed in the machines by seven conductor cables made up of six copper and one advance wire.

Helm angle indicator.

Time clock.

The center panel carries four red lights near the top, two on each side, which indicate which direct current feeders from the engine room are alive. It also carries a red, green and white light on each side of the lower part of the panel to give the operator information on the operation of the turbine power limit and condition of throttle. The red light indicates that the governor has opened the governor-controlled valve as far as the power limit stop will allow. The green light indicates that the motor operating the power limit is in operation. The white light indicates that the main throttle is open.

The main control board also carries the supply switches to the motor ventilating blower, the liquid rheostat pumps, and the lubricating oil relay pumps. Each switch to the motor ventilating blowers and lubricating oil relay pumps has a red indicating light which shows when voltage is supplied to the commutator brushes of the motors.

The liquid rheostat, with its electrolyte pumps and controllers, has been described under the main motors.

The control room is the "central" of the engineer department as regards communication systems. The following systems are installed:

1. Electric engine order telegraph, starboard and port, from the bridge, conning tower, or central station.
2. Mechanical engine order telegraph, starboard and port, from the central station.
3. Electric revolution telegraph, starboard and port, from the bridge, conning tower, or central station.
4. Ship's service telephone.
5. Fire-control telephone to bridge, conning tower, central station and steering room.
6. Fire-control telephone circuit to all firerooms and engine rooms.
7. Loud speaking telephone receiver.
8. Callophone, transmitting and receiving to bridge, conning tower and central station.
9. Callophone, transmitting and receiving to "Battle II."
10. Telautograph to all main machinery compartments and Engineer's office.
11. Voice tubes to all important stations, such as, bridge, conning tower, central station, all main machinery compartments, distribution rooms, steering room, etc. All voice tubes are fitted with call bells and annunciators.
12. General alarm and cease firing gongs.

A full description of the fireroom control station with method of operating has been given in the previous issue of the JOURNAL. This being the first vessel to be commissioned with each boiler in a separate watertight compartment, some efficient means of boiler control had to be provided. The system as installed has operated perfectly from the first.

## FORCED LUBRICATING SYSTEM.

The lubrication of all main turbine, generator and motor bearings, is effected by a continuous circulation of oil through a system of delivery and return pipes. There are two independent systems, one for the forward engine room and the other for the after engine room which includes the main motor and thrust bearings. Each system consists generally of one motor-driven and one steam-driven oil pump, suction and discharge strainers, oil coolers, oil cooler circulation pump, drain and stowage tanks, and a centrifugal oil purifier. The after system has 3 additional motor-driven oil relay pumps for returning from the control room, motor rooms and thrust block rooms drainage oil to the engine room drain tank.

The pumps draw oil from the drain tanks, through suction strainers, and delivers it at 80 pounds per square inch pressure. The oil after passing through the coolers branches into two lines, one to the hydraulic system for speed control and the other through a special constant pressure valve and a baffler valve in parallel to the lubricating system. The special constant pressure valve is designed to maintain a pressure of 80 pounds on the hydraulic system for speed control, all excess over the requirement being passed to the lubricating system. The baffler valve is installed in order to safeguard the lubricating system so that in case of failure of the constant pressure valve some oil will be passed to this system. A portion of the oil after being used in the speed control system is exhausted to the lubricating oil system and used there. A special constant pressure valve is fitted in the lubricating oil system to maintain pressure at 10 pounds, excess oil supplied by the pump being returned to the oil drain tank. Thermometers are placed in the inlet and outlet to the oil cooler and to the outlet of each bearing. There is also fitted a sight discharge at each bearing.

PUMP DATA.

Type—Kinney rotary plunger. Two direct connected motor driven and two geared steam driven.

| | |
|---|---|
| Capacity, gallons per minute | 250 |
| Head, feet | 185 |
| Speed, R.P.M. | 600 |
| Suction, diameter, inches | 4 |
| Discharge, diameter, inches | 4 |
| Motor, horsepower | 25 |
| voltage | 240 |
| speed, R.P.M. | 400/600 |
| Turbine horsepower | 25 |
| speed, R.P.M. | 2,400 |

The turbine is fitted with hydraulic pressure governor, constant speed governor, emergency overspeed trip, and special automatic valve for cutting the turbine-driven unit in service in case of failure of the motor-driven unit.

OIL COOLER CIRCULATING PUMP DATA.

Name of manufacturer—Westinghouse Electric and Manufacturing Company.

Type—Variable speed, motor driven, direct connected, horizontal, single stage, single impeller.

| | |
|---|---|
| Number (1 each engine room) | 2 |
| Capacity, G.P.M. | 300 |
| Head, feet | 115 |
| Speed, R.P.M. | 1,700 |
| Suction, diameter, inches | 4 |
| Discharge, diameter, inches (oil cooler and fire main for flushing) | 3 |
| Motor, horsepower | 16 |
| speed, R.P.M. | 583/1,700 |
| voltage | 240 |

LUBRICATING OIL RELAY PUMP DATA.

Name of manufacturer—Wm. E. Quimby, Inc.

Type—Quimby screw, horizontal, constant speed, motor driven.

| | |
|---|---|
| Number | 3 |
| Capacity, G.P.M. | 65 |
| Speed, R.P.M. | 1,325 |
| Head, feet | 100 |
| Motor, horsepower | 7½ |
| voltage | 240 |

type, Electro-Dynamic Company.

### OIL COOLER DATA.

Name of manufacturer—Schutte & Koerting Company.
Number (two in each engine room) ..... 4
Type, two No. 5 Zimmerman, two No. 6 Zimmerman.

| | | |
|---|---|---|
| Cooling surface, square feet, | No. 5 | 127 |
| | No. 6 | 170 |
| Diameter, inside, inches, | No. 5 | 12 7/8 |
| | No. 6 | 14 7/8 |
| Tubes, number | No. 5 | 241 |
| | No. 6 | 297 |
| diameter, outside, inch | | 9/16 |
| thickness, inch | No. 5 | .032 |
| | No. 6 | .049 |
| length, between tube sheets | | |
| | No. 5 | 46 |
| | No. 6 | 50 |

### CENTRIFUGAL, OIL, PURIFIER DATA.

Name of manufacturer—De Laval Turbine Equipment Company. Type, motor driven, centrifugal.

| | |
|---|---|
| Number (one in each engine room) | 2 |
| Capacity, gallons per hour | 100 |
| Motors, type, Diehl. | |
| horsepower | 1 |
| speed, R.P.M. | 1,750 |

### STOWAGE AND DRAIN TANK DATA.

| | | |
|---|---|---|
| Stowage tanks, number | | 3 |
| capacity, each, gallons | | 1,005 |
| location, center motor room. | | |
| Drain tanks, number | | 4 |
| capacity, forward engine room | | 851 |
| after engine room | | 1,218 |
| control room, gallons, one | | 100 |
| and one | | 200 |
| Treating tanks, number | | 2 |
| capacity, gallons, forward engine room | | 842 |
| capacity, gallons, after engine room | | 1,173 |

### WATER SERVICE SYSTEM.

The entire water service for the thrust bearings, liquid rheostats, spring bearings, stern tubes, and auxiliary machinery bearings fitted for cooling, is supplied from connections taken

off the main circulating pumps, oil cooler circulating pumps, and fire and bilge pumps discharges.

The forward and after engine room water service is entirely independent, the after engine room system includes the control room, motor rooms and shaft alleys. Branches controlled by separate valves or cocks are led to the various bearings requiring cooling. Hose valves and universal pipe connections are also provided at required locations. Drains are connected to a closed circuit and discharge overboard, the after system through the after auxiliary overboard sea chest and the forward system through the forward chest.

## BOILERS.

There are eight Babcock and Wilcox water tube boilers, each located in a separate watertight compartment.

The boilers are equipped for burning fuel oil, each boiler having six burners of the Peabody Mechanical Atomizer type. They have a total heating surface of 41,768 square feet, exclusive of superheating surface, and are designed with sufficient power to operate the entire machinery plant at full power.

A great deal of trouble had been experienced by the oxidizing of the brick work retaining bolts. After thorough investigation of the subject "calorized" bolts were fitted. There were also installed a number of monel metal bolts for experiment.

Due to deflection of the fire tubes on boilers of similar type on former ships at full power, an internal feed pipe was installed for discharging the water directly into the down-comers. Near the point of discharge to the down-comers a valve was placed in this line with provision for outside operation so they could be opened or closed as desired. At ordinary powers the feed water is discharged through an internal feed pipe which extends the length of the drum, the openings being on the top of the pipe. The use of the discharge into the down-comers continuously would cause rapid corrosion of the tubes and down-comer.

Uptakes are provided, the four forward uniting into a forward smoke pipe and the four after into an after smoke pipe. The uptake from each boiler is provided with a pyrometer for indicating the temperature and with connection for obtaining a sample of gas for analysis. The uptakes and smoke pipes are designed to give a clear ratio in square feet to the weight in pounds of the oil burned per hour at full power of 1 to 300, except through armor bars where the ratio is increased to about 1 to 250. The uptakes and smoke pipes are properly insulated by air spaces or the use of approved insulating material. The smoke pipe has an area of 63.83 square feet and 93 feet 3 inches in height above the center line of the oil burners. The interior of the smoke pipes are divided into two vertical discharge passages, having air space between and free to expand separately.

BOILER DATA:

| | |
|---|---|
| Number | 8 |
| Pressure, working, pounds per square inch | 280 |
| test, pounds per square inch, water | 420 |
| test, pounds per square inch, steam | 330 |
| Heighth to top, external, feet and inches | 14– 8 1/2 |
| Length on floor, feet and inches | 8– 1 7/8 |
| Width on floor, feet and inches | 17– 7 1/4 |
| Drum, diameter, inside, inches | 36 |
| length, feet and inches | 18– 7 1/4 |
| thickness, inch | 29/32 |
| manholes, inches | 11×15 |
| Number of tube headers, each boiler | 58 |
| Distance between headers, feet and inches | 8– 0 |
| Number of tubes, each boiler | |
| 2-inch generating tubes, 134 mils thick, 8 feet 2 inches long | 1,140 |
| 4-inch generating tubes, 165 mils thick, 7 feet 4½ inches long | 29 |
| 4-inch downcomer tubes, 165 mils thick, 6 feet 11½ inches long | 2 |
| 2-inch "U" superheater tubes, 165 mils thick | 152 |
| Number of furnaces, each boiler | 1 |
| Furnace volume, each boiler, cubic feet | 440 |

| | |
|---|---|
| Heating surface, each boiler, square feet | 5,221 |
| Superheating surface, each boiler, square feet | 521 |
| Kind of forced draft | closed fireroom |
| Diameter of main steam stop valve, inches | 7 |
| forward and after main feed stop and check valves, inches | 3 |
| surface-blow valves, inches | 01 1/2 |
| bottom-blow valves (2), inches | 01 1/2 |
| safety-valve (2), duplex, inches | 04 |

BOILER FEED SYSTEM.

The boiler feed system consists generally as follows:

Reserve feed tanks and connections.

Main feed and filter tanks.

Main feed pumps.

Reserve feed pump.

Feed water heaters.

Necessary piping system.

There are five reserve feed tanks, with capacities in gallons as follows:

A-8, 38,190; A-9, 26,000; C-2, 26,640; C-3, 34,790; D-1, 32,580. Total, 158,200 gallons.

Connections are fitted in the ship's sides, for hose connections, and connected to the combined filling and suction manifold in each of the machinery rooms. The distiller fresh water main also discharges into these manifolds. The reserve feed pump in each machinery room is arranged to draw from any tank or to discharge to any other tank.

One feed and filter tank of 4,570 gallons capacity is installed in each engine room. A horizontal partition divides the tank into two parts, the upper part for the filter and the lower part being the feed tank proper. The capacity of the two parts is, respectively, 830 gallons and 3,740 gallons. Each tank is provided with the following principal connections:

10-inch main air pump discharge.

6-inch dynamo air pump and dynamo air separating tank discharge.

2½-inch high pressure drain line.
5-inch drain from main air separating tanks.
2½-inch drain from evaporating room.
2-inch drain from fuel oil inspection tank.
2½-inch low pressure drain line.
1½-inch drain to bilge.
3½-inch feed heater drain line.
3½-inch discharge from main feed pump relief valve.
2-inch main feed by-pass.
2½-inch re-circulating pipe to main condenser.
2-inch vapor pipe drain.
8-inch overflow.
2-inch overflow from reserve feed tank.
10-inch cross-connection and pump suction.
2-inch vapor from each filter compartment (four).
3-inch vapor from feed compartment.

There are two Cameron direct connected, turbine-driven, horizontal, 4-stage centrifugal main feed pumps installed in each engine room. Each has a capacity of 500 gallons per minute, against 920 foot head, at 3,600 R.P.M. The turbine is fitted with the Ideal hydraulic speed regulating governor for maintaining constant pressure, a constant speed governor, and an emergency trip overspeed governor. It is 190-horse-power, single stage, and manufactured by the General Electric Company.

A reserve feed pump is installed in each engine room. It is a M. T. Davidson vertical, piston, double acting, single type and size 12×8×12.

One Griscom-Russell Company's Reilly multicoil feed water heater is installed in each engine room. Each heater has a total heating surface of 430 square feet and a rated capacity of 240,000 pounds of water from 80 degrees to 205 degrees Fahrenheit with steam at 10 pounds pressure gauge. The heating agent is auxiliary exhaust steam and the drains from the high pressure drain system.

The main feed pumps discharge the feed water via the feed heaters, or by-pass same if desired, to the boilers. The forward and after engine room feed system consists of two independent systems, each connecting separately to each boiler. The discharges from the two main feed pumps and the reserve feed pump combine to form a common line. Relief valves are fitted at the pumps to prevent damage to the pump casings. A by-pass from the pressure line to the feed tank is provided in order to relieve the pump when working against closed checks on all boilers. In order to prevent the main feed pumps from becoming vapor bound, a ¾-inch pipe is connected to the highest point of the suction casing and carried above the level of the water in the feed tank. Control of the feed water into the boiler is by a 3-inch feed-check fitted to each system at the boiler drum. Cut-out valves for each boiler are fitted, to be operated from the engine room.

## FUEL OIL SYSTEM.

Approximately 4,656 tons of fuel oil may be stored in 110 compartments, arranged, primarily, with a view of obtaining maximum torpedo protection. Four of these compartments are used as service tanks, two under each machinery room. Each compartment is fitted with an air escape and blowpipe for sounding connection, air escape and blow line.

For the purpose of ascertaining the height and weight of oil in each tank at any time, a pneumercator system is installed. An annunciator is fitted on each pneumercator board to give visible and audible warning when the tank is filled to 95 per cent of its full capacity. The pneumercators for the stowage tanks are located in groups of three and four, protected by screened and locked box, on the third deck. Those for the four service bottoms are located in each machinery room.

Oil is taken on board through eight standard flanged 6-inch filling hose connections. These connections are combined and lead to two relay tanks, located on the third deck, which

strain the oil and distributes it through suitable piping to any of the stowage or service tanks. Two four-inch emergency connections are provided, leading to the fuel oil booster pump which distributes it to any of the stowage or service tanks.

Each inboard stowage tank and service tank is provided with high and low suction and with steam heating coil for use in handling oil during cold weather.

One light-service single booster pump, with suction connections to fuel-oil main cross-connection and service tanks, and with discharge to the service tanks, fuel oil main, filling line to galley tank and for steamers, fuel oil relay tank, and fuel oil service pump suction, is installed in each engine room. These pumps are M. T. Davidson vertical, piston, double acting, single type and size 6×8×12.

There are two heavy pressure, horizontal, turbo-driven, reduction gear, rotary service pumps of the Kinney type installed in each engine room. Ideal hydraulic pressure regulators are provided to control the speed of the turbine in order to maintain a constant pressure. The turbines are also provided with constant speed governors and with centrifugal emergency trip governors set to trip 20 per cent above the setting of the constant speed governors in order to prevent excessive overspeeding. They have a 3-inch suction, through a duplex suction strainer, to the fuel oil distributing manifold and to the discharge from the booster pump. The pump discharge to the burners is through a duplex discharge strainer, oil meter, fuel oil heaters, and through distributing pipe lines. In the pump side of the discharge valve is fitted a spring loaded relief valve and also a by-pass valve to the pump suction for use when only a small quantity of oil is being used. Each pump is provided with an air chamber on its discharge for insuring constant pressure. The air chambers are connected to the pneumatic main and fitted with test cocks.

The oil meters are of the Empire type, and are arranged so they can be by-passed.

The oil heaters are of the Schutte and Koerting pressure type, each of sufficient capacity to heat 16,000 pounds of oil from 70 degrees to 240 degrees Fahrenheit with steam at 265 pounds pressure. They have 43.08 square feet heating surface, are fitted with proper drains, and by-pass is provided for emergency use.

For raising steam with no pressure available for the operation of the pump, a 1½×3½-inch duplex hand pump is fitted in each engine room, capable of supplying oil at 200 pounds pressure to two burners.

Six atomizers for each boiler are carried in one row on the boiler frot, each burner is fitted with stop valve for individual control. They are of the Peabody mechanical atomizing type. The master control valve to each boiler supply is arranged to work from the adjacent engine room as well as the fireroom.

The forward and after fuel oil pressure lines are continuous, representing a closed system fitted with proper segregating valves. All joints and fittings in the pressure piping are made up without the use of gaskets.

## FORCED DRAFT BLOWERS.

Each fireroom has installed a forced draft blower capable of supplying continuously sufficient air for the boiler to burn smokeless, the amount of oil required for maximum power. They are located in specially constructed blower rooms just below the third deck and above the working platform in front of the boilers.

The blowers are of the B. F. Sturtevant, horizontal, double inlet, direct connected, turbine driven type and each is rated at 24,000 cubic feet of air per minute, at 8 inches air pressure and 1,800 R.P.M. It has been necessary to increase the operating speed for full power to 2,000 R.P.M. in order to obtain the necessary air. This is probably due to the restricted

FIG. 33.—GENERAL VIEW OF CONTROL ROOM.

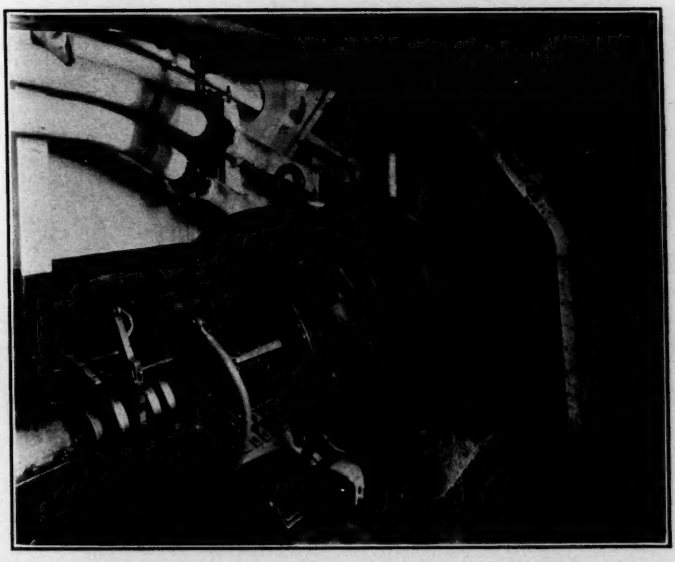


FIG. 34.—300-K.W. TURBO GENERATING SET AS INSTALLED.

inlet ducts. The stop valve to each blower is arranged to operate from the blower room and the fire room. The connection in the fire room is arranged to give a small movement of the valve for a large movement of the operating hand wheel to secure close regulation.

ELECTRIC PLANT.

As mentioned under main generator excitation, there are three 300 K.W. direct current, 120/240-volt, three wire, reduction gear 6,000 to 900 R.P.M., turbo-generating sets, installed in each engine room. The sets in the same engine room are arranged for parallel operation for ship's light and power. The forward engine room and after engine room sets cannot be paralleled. Figure 34 shows one of these sets as installed in the engine room.

The non-condensing turbine is of the impulse type and designed to operate on 250 pounds steam pressure with 10 pounds back pressure, while the two other turbines are of the impulse reaction type and designed to operate on 200 pounds steam pressure and 25 inches vacuum. Both turbines are fitted with hand-operated nozzles for overload condition and have constant speed governors for operating both primary and secondary valves, emergency overspeed trip, relief valve on casing to limit the pressure to 50 pounds with wide open throttle and closed exhaust.

Each turbine is fitted with a forced lubrication oil pump driven from an extension of the governor spindle. This pump is located in the base of the gear casing which also serves as an oil reservoir. The oil is delivered to the bearings and gears through outside oil coolers.

The generators are compound wound and designed to deliver 1/3 overload for two hours. Brass collector rings are mounted on the armature shafts which are connected to compensators or balance coils. When used with these compensators the generators, operating at 240 volts with an unbalanced

load of 43 per cent will maintain a voltage balance of not less than 117 volts and not more than 123 volts between the neutral and outside wires. Shunt field rheostats for each generator are mounted on the auxiliary switchboard. All generators are flat compounded within one volt either way from 240 volts normal and full load points.

The two condensing turgo-generating sets in each engine room have an independent condensing plant located below the generators. Each condensing plant consists of one condenser, air pump, circulating pump, condensate pump, and air ejectors. This condenser also has connections to the auxiliary exhaust line, low pressure drain line and evaporator coil drain. The air pumps discharge directly to the feed and filter tank and the condensate pump and ejectors to the feed and filter tank through air separating tank.

DYNAMO CONDENSER DATA.

Type—Lovekin. The construction and general arrangement is the same as for the main condenser.

| | |
|---|---|
| Cooling surface, square feet | 1,925 |
| Diameter, inside, feet and inches | 4– 6 |
| Length, between tube sheets, feet and inches | 7– 8 |
| Tubes, number | 1,534 |
| Openings, exhaust, diameter, inches | 18 |
| air pump suction diameter, inches | 7 |
| ejector, suction, diameter, inches | 4 |
| circulating water inlet, diameter, inches | 8 |
| circulating water outlet, diameter, inches | 6 |

DYNAMO CIRCULATING PUMP DATA.

Type—Horizontal, variable speed, motor driven, centrifugal, single stage and single runner.

Manufacturer—Westinghouse Electric and Manufacturing Company.

| | |
|---|---|
| Capacity, G.P.M. | 2,000 |
| Head, feet | 20 |
| Speed, R.P.M. | 700 |
| Suction, diameter, inches | 12 |
| Discharge, diameter, inches | 10 |
| Motor, horsepower | 17 |

| | |
|---|---|
| Speed, R.P.M. | 350/200 |
| Voltage | 240 |

Controller. Panel mounted, hand operated sliding arm starting and field regulating rheostat, circuit breaker with overload and no voltage trip.

## CONDENSATE PUMP DATA.

Type. Vertical, centrifugal, variable speed, motor driven, single stage and single runner.

Manufacturer—Westinghouse Electric and Manufacturing Company.

| | |
|---|---|
| Capacity, G. P. M. | 75 |
| Head, feet | 70 |
| Speed, R.P.M. | 1,700 |
| Suction, diameter, inches | 5 |
| Discharge, diameter, inches | 3 |
| Motor, horsepower | 4 |
| speed, R.P.M. | 1,500/1,800 |
| voltage | 240 |

## DYNAMO AIR PUMP.

Type. M. T. Davidson, vertical, twin air cylinders, single steam cylinder.

| | |
|---|---|
| Size, inches | 9×18×18×12 |
| Suction, diameter, inches | 7 |
| Discharge, diameter, inches | 6 |

## AIR EJECTOR DATA.

Type—Westinghouse LeBlanc. Three two-stage ejectors are mounted in parallel.

| | |
|---|---|
| Capacity, pounds free air per hour | 18 |
| Vacuum referred to 30 inches barometer | 28 |
| Steam pressure, pounds gage | 125 |
| Steam per hour, pounds | 440 |

## DYNAMO AIR SEPARATING TANKS.

Construction and general arrangement the same as for the main air separating tanks.

| | |
|---|---|
| Capacity, pounds steam per hour | 1,980 |
| pounds condensate per hour | 110,000 |
| Cooling surface, square feet | 35 |
| water, G.P.M. | 100 |

### FIRE AND BILGE PUMPS.

There are four fire and bilge pumps, two located in each engine room. They are M. T. Davidson vertical, piston, double-acting, single and have the following suction connections, 10-inch from the sea, 8-inch drainage, 1½-inch hose and 8-inch portable fuel oil pumping out. The following discharge connections are provided: 8-inch fire main, 8-inch overboard, 2½-inch hose and 4-inch oil cooler and water service. The pump size is 16 × 14 × 24.

### STEAM LINES.

Figure 35 shows the arrangement of the steam lines. It can be seen from this illustration the simplification in pipe arrangements which is gained by the use of electric drive.

### AUXILIARY EXHAUST PIPING.

In the engine room is installed an auxiliary exhaust piping system, connected to all of the auxiliary machinery. It is fitted with valves to direct the exhaust steam into the main condensers, dynamo condensers, feed water heaters, main turbine, evaporators, the atmosphere escape. The connections to the dynamo condenser, and main turbine are fitted with constant back pressure valves for maintaining the pressure in the line constant. By regulating the setting of these valves the exhaust will be directed into these units in any order desired. The connection to the non-condensing 300 K.W. turbo-generating set also has a constant back pressure valve installed.

### ESCAPE PIPES AND ATMOSPHERIC EXHAUST.

Branches leading from the safety valves, 300 K.W. turbine relief valves, evaporator steam supply relief valves, atmospheric exhaust line, top of feed tanks, and the main and dynamo air separating tanks are led to an escape pipe located on each smoke pipe.

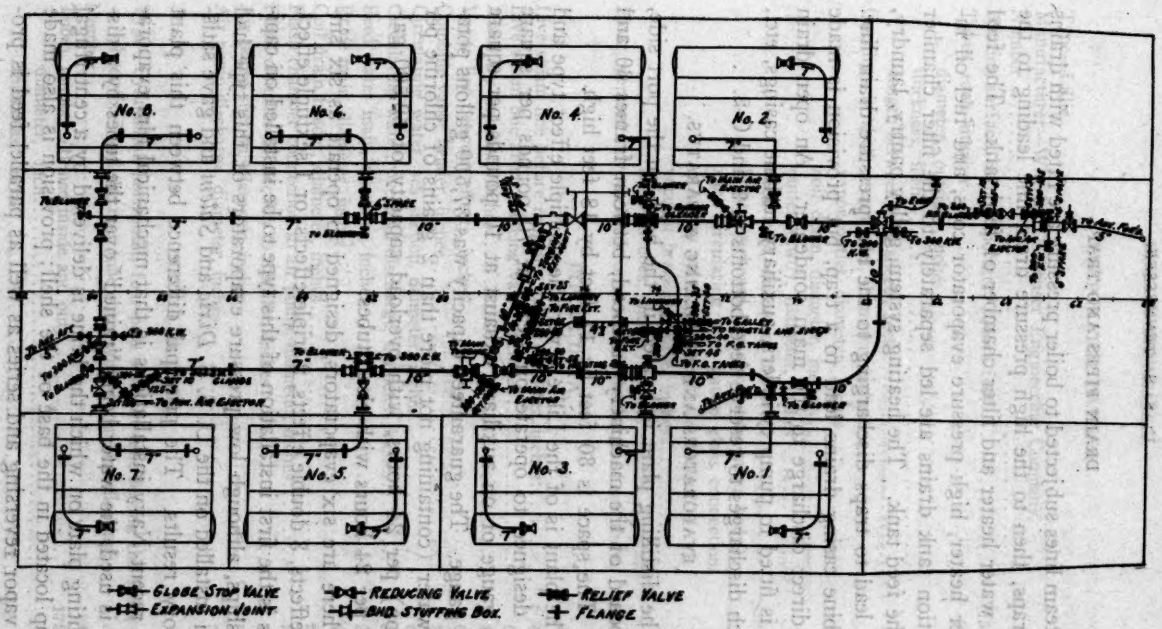



FIG. 35.—DIAGRAM OF STEAM LINES.

## DRAIN PIPES AND TRAPS.

Steam lines subjected to boiler pressure are fitted with drains to traps, then to the high pressure drain line leading to the feed water heater and filter chamber of feed tank. The feed water heater, high pressure evaporator coils, and fuel oil inspection tank drains are led separately to the filter chamber of the feed tank. The heating system, galley pantry, laundry, etc., lead to traps discharging to the low pressure drain line. Turbine casing drains lead to a trap, but provision is made for direct discharge to the main condenser. An open drain line is fitted to pump cylinders, auxiliary turbine casings, etc., which discharges to reserve feed bottoms A-9 and C-2.

## EVAPORATORS AND DISTILLING APPARATUS.

The distilling plant is located in the hold, on the port side, outboard of the machinery bulkhead, between frames 60 and 68. The space is 30 feet by 17 feet by 18 feet high.

The plant is of the Lillie reversible multiple effect type and was designed to operate on steam at 70 pounds per square inch gauge or on auxiliary exhaust at 10 pounds per square inch gauge. The guaranteed capacity was 37,500 gallons potable water (containing not more than 2 grains of chlorine per gallon) per 24 hours, with overload capacity of 52,500 gallons per 24 hours with clean tubes.

There are six evaporators, designed to operate as six single effects, 3 double effects, 2 triple effects, or 1 sextuple effect. This is the first installation of this type to be installed on capital ships, although low pressure evaporators of this type had been installed on the U. S. S. *Dixie* and *Salem* and gave satisfactory results. The principal difference between this plant and recent Navy installations is that mechanical film evaporation is used, the water being "rained" over the tubes by a distributing plate on which the brine is delivered by a centrifugal pump located in the base of the shell; provision is also made for vapor reversing and series as well as parallel feed is provided.

## EVAPORATOR DATA.

Type—Lillie.
Manufacturer—Wheeler Condenser Engineering Company.

| | |
|---|---|
| Number | 6 |
| Heating surface, each evaporator, square feet | 150 |
| Tubes, each, number | 83 |
| diameter, outside, inches | 1½ |
| thickness, inch | .065 |
| length between tube sheets, feet | 5 |
| Openings in shell, steam inlet, diameter, inches | 6 |
| vapor outlet, diameter, inches | 6 |
| coil drain, diameter, inches | 1½ |
| outlet (two), diameter, inches | 1½ |
| feed inlet, diameter, inches | 1½ |
| brine overboard (on circulating pump body) | 2 |
| brine circulating inlet gauge glass connections | |
| safety valve on steam head, inches | 1 |
| vacuum breaking valve, inches | 1 |
| air vent cock, inch | ½ |

Float operated valve in head for automatic discharge of coil drain.

## PIPING DATA.

| | |
|---|---|
| Live steam line before reducing valve, diameter, inches | 4½ |
| Live steam line after reducing valve, inches | 6 |
| Auxiliary exhaust, inches | 8 |
| Vapor line between effects, inches | 6 |
| Combine vapor header, inches | 8 |
| Circulating water, sea suction, diameter, inches | 6 |
| Circulating water, overload discharge, diameter, inches | 5½ |
| Feed water main, diameter, inches | 2½ |
| Brine main, diameter, inches | 2 |
| Coil drain header, diameter, inches | 2½ |
| Flash chamber connection to air pump, diameter, inches | 2½ |
| Flash chamber connection to distiller, inches | 4 |
| Air pump suction, diameter, inches | 6 |
| Air pump discharge, diameter, inches | 5 |

## DISTILLER DATA.

Type—Cylindrical, horizontal, four-pass straight tube having the front tube sheet fixed and the rear tube sheet with water head floating.

| | |
|---|---|
| Number | 4 |
| Tubes, number (tinned inside and out) | 172 |
| diameter, inch | ⅝ |
| thickness, inch | .065 |
| length, feet and inches | 5–4¼ |

| | |
|---|---|
| Cooling surface, each, square feet | 150 |
| Connections, vapor inlet, diameter, inches | 6 |
| air and condensate outlet, diameter, inches | 4 |
| circulating water inlet, diameter, inches | 3 |
| circulating water outlet, diameter, inches | 3 |
| air cock on water chest, inch | ½ |

COIL DRAIN FEED WATER HEATER DATA.

| | |
|---|---|
| Type. Four pass, counter current, U tubes. | |
| Number | 1 |
| Tubes, number | 52 |
| diameter, outside, inch | ⅝ |
| thickness, inch | .049 |
| Heating surface, square feet | 46 |
| Coil drain inlet, outlet, feed inlet, feed outlet, diameter, inches | 2½ |

FRESH WATER METER DATA.

| | |
|---|---|
| Type. Cochrane Precision Liquid Meter. | |
| Number | 1 |
| Capacity per hour in gallons | 18,500 |
| Inlet, diameter, inches | 2½ |
| Outlet, diameter, inches | 3 |

DISTILLER CIRCULATING PUMP DATA.

| | |
|---|---|
| Type. Wheeler horizontal centrifugal pump with Terry turbine. | |
| Number | 2 |
| Capacity, gallons per minute | 350 |
| Head, feet | 40 |
| Speed, R.P.M. | 1,800 |
| Connections, steam, inches | 2 |
| exhaust, inches | 4 |
| suction, inches | 4 |
| discharge, inches | 4 |

AIR PUMP DATA.

| | |
|---|---|
| Type. Worthington featherweight, vertical, simplex, double acting. | |
| Number | 2 |
| Size, inches | 7½×14×12 |
| Connections, steam, inches | 1 |
| exhaust, inches | 1½ |
| suction, inches | 6 |
| discharge, inches | 5 |

FRESH WATER PUMP DATA.

| | |
|---|---|
| Type. M. T. Davidson vertical, simplex, double acting, piston pumps. | |
| Number | 2 |
| Size, inches | 4½×5×6 |
| Connections, steam, inch | ½ |
| exhaust, inch | ¾ |
| suction, inches | 2½ |
| discharge, inches | 2 |

FEED, BRINE AND RESERVE PUMP DATA.

| | |
|---|---|
| Type. George F. Blake vertical, simplex, double acting, piston pumps. | |
| Number | 3 |
| Size, inches | 4½×6×6 |
| Connections, steam, inch | ½ |
| exhaust, inch | ¾ |
| suction, inches | 2½ |
| discharge, inches | 2 |

The reserve pump is so connected that it can be used for either brine overboard or feed.

BRINE CIRCULATING PUMP DATA.

| | |
|---|---|
| Type. Centrifugal, motor driven. | |
| Number, pumps | 6 |
| motors | 2 |
| Motor, type, Westinghouse. | |
| horsepower | 15 |
| voltage | 230 |
| speed, R.P.M. | 1,000 |

Both motors and all six pumps for circulating the water in the shells are mounted in line and connected by flexible couplings. One motor is used to drive all pumps.

In the air pump suction are fitted cells for measuring continuously the grains of chlorine per gallon. The instrument was manufactured by the Leeds-Northrop Company and furnished by the Babcock and Wilcox Company. It consists of a small rotary convertor for supplying A.C. voltage to the cells and a chart recording instrument for measuring the change of resistance across the cell terminals which gives a direct indication of the chlorine per gallon. In connection with the

instrument are contacts which causes a white light to burn in the evaporator room when the chlorine per gallon is below .5, a green light between .5 and 5, and a red light for over 5. The operator has instructions to put only "white light water" in reserve feed bottoms, "green light water" in ship's tanks, and "red light water" in the bilge. In addition to this there are fitted cells in each of the evaporator brine discharge pipes from which the concentration of the brine can be obtained at any time by hand operation of an instrument.

A great deal of trouble was experienced in the operation of the plant on high pressure steam due to severe sulphate scale incrustations. Some of the tubes have been removed from the high pressure evaporators in order to facilitate the removal of scale. It was determined that the rapidity of the scale formation was a direct function of the temperature and brine concentration. This trouble is experienced in all evaporating plants where high temperatures are used. The plant is now operated only on low pressure, in double effect, without vapor reversing, with parallel feed, and carry the brine concentration from two to three, thirty seconds. Under this condition the plant is giving most excellent results. Below is given some operating data, under actual service condition. The data represents average results over one week's operation in port and for two days at sea. The data was taken after 45 days' continuous operation without cleaning or scaling.

REFRIGERATING SYSTEM.

The ice machine room is situated between frames No. 60 and 68, on the starboard side of the first platform deck, between the fifth torpedo bulkhead and the machinery space bulkhead.

The plant consists of three independent ice machines (York Ice Machine Company) driven by General Electric direct current shunt wound electric motors of 10 H.P. each, 230 volts, 200-300 R.P.M.; two direct current General Electric shunt

## AVERAGE PRESSURE.

| A | | B | | C | | D | | E | | F | | | | | Temperatures. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Coil. | Shell. | Coil. | Shell. | Coil. | Shell. | Coil. | Shell. | Coil. | Shell. | Coil. | Shell. | Aux. Steam. | Vacuum at pump. | Water distilled per 24 hours. | Chlorine grains per gallon. | Injection. | Overboard Discharge. | Feed heater in. | Feed heater out. | Remarks. |
| | *ins.* | *ins.* | *ins.* | | *ins.* | *ins.* | *ins.* | | *ins.* | *ins.* | *ins.* | | *ins.* | | | | | | | |
| 0 | 17 | 17 | 25 | 0 | 18 | 18 | 24 | 1 | 16 | 16 | 24 | 2.4 | 26.5 | 18,000 | 2/5 | 62 | 78 | 78 | 104 | In Port. |
| 2 | 15 | 15 | 25 | 3.5 | 16 | 16 | 24 | 2.5 | 15 | 15 | 23 | 5 | 25 | 24.000 | 2/5 | 60 | 84 | 84 | 116 | At Sea. |

*Conditions of Operation.*—3 double effects, 1 air pump, 1 brine pump, 1 fresh water pump, 1 distiller circulating pump, 1 brine circulating pump motor, brine overboard 2/32, parallel feed from distiller circulating pump discharge and through coil drain heater, 2 men on watch.

wound motors of 3 H.P. each, 230 volts, 1,800 R.P.M. driving centrifugal pumps for circulating the brine; two direct current General Electric shunt wound motors of 2 H.P. each, 230 volts, 1,800 R.P.M. for circulating the seawater through the condensers. The capacity of the brine pump is 48 G.P.M. of brine Sp. Gr. 1.25, against a total head of 70 feet. The capacity of the circulating pump is 42 G.P.M. of sea water, against a total head of 46 feet. The above capacities are per pump. There are two brine coolers; three units making up the $CO_2$ condenser; one brine storage; two ice-making boxes and ice-can thawing tank; two scuttle butts, located on main and second decks; cold storage rooms for meats, officers' mess, eggs and cheese, and cream and butter, etc., located on the third deck; and a 40-quart ice-cream freezer and a hardening cabinet, located on main deck. There is storage space for ten $CO_2$ flasks in the ice-machine room, and for six additional flasks in hatchway leading down to ice-machine room from the deck above. The ice-making box is equipped with an overhead trolley for handling the ice cans. The thawing tank is placed at the after end of the ice-making box.

### GENERAL WORKSHOP.

Installed in the general workshop, located on the third deck over the forward engine room, are the following tools. All tools are driven by separate 230-volt electric motors.

| *No.* | *Description.* | *Motor Data.* |
|---|---|---|
| 1 | 25-inch by 50-inch swing LeBlond screw cutting, back geared, extension gap lathe. | 10 H.P. Reliance 500/1,500 R.P.M. |
| 1 | American Tool Works screw cutting, all geared head toolroom lathe, 16½-inch swing, taking 40 inches between centers. | 3 H.P. Reliance 1,500 R.P.M. |
| 1 | American Tool Works screw cutting, back geared, toolroom lathe, to swing 16½ inches over the ways and 64 inches between centers. | 3 H.P. Reliance 1,150 R.P.M. |
| 1 | Rivet Lathe and Grinder Company's precision lathe, 8-inch swing by 18 inches between centers. | |

| | |
|---|---|
| 1—American Tool Works column tool-shaper, 24-inch stroke and 24-inch traverse, with extension base. | 3 H.P. Reliance 1,150 R.P.M. |
| 1—American Tool Works power-feed, radial drill, to drill up to 1½-inch diameter. | 3 H.P. Reliance 1,150 R.P.M. |
| 1—Sipp Machine Company 16-inch high speed sensitive drill. | General Electric 3 H.P. 1,100/1,150 R.P.M. |
| 1—Keaney-Trecker Company's universal milling machine, 24-inch longitudinal feed, 14-inch vertical movement and 6-inch traverse, vertical feed attachment. | General Electric 3 H.P. 1,150 R.P.M. |
| 1—Willey, 2-wheel emery grinder on column. | Reliance 2 H.P. |
| 1—Willey hacksaw, 4-inch capacity. | |
| 1—Cincinnati Electric Tool Company's portable surface grinder, Type TT. | 1 H.P. |
| 1—Pneumatic tool drive portable cylinder boring machine. | |
| 1—Cincinnati Electric Tool Company's portable drill to drill up to ½-inch diameter. | |
| 1—Cast iron laying-out table fitted with T slots for securing work. | |
| Work benches and machinists vises are fitted to suit the space available. | |

The blacksmith shop, foundry, and pipe and coppersmith's shop are located together in a compartment on the main deck, amidships.

The blacksmith shop is provided with one permanent and one portable forge, together with anvil and necessary tools. The permanent forge is of the oil-burning type and the portable forge of the folding coal-burning type.

The foundry is equipped with one marine oil-burning tilting crucible furnace. Necessary tools, crucibles and equipment are furnished for carrying on the foundry work.

The pipe and coppersmith shop is provided with the following epuipment.

1—Motor-driven pipe cutting and threading machine, to cut and thread pipe from 3 to 6 inches.

1—Pipe bending slab.

1—Portable oxyacetylene outfit, Milburn type.

1—Oil forge "Houck" No. 4.
1—Tinners' combine shear and punch.
1—Tinners' slip-roll forming machine.
1—Tinners' leading machine with rotary stand.
1—Tinners' conical brake.

The combined shops are fitted with necessary work benches, vises, etc.

## STORE ROOM AND STORE SYSTEM.

Due to troubles experienced in the past in stowing, accounting for, and issue of machinery and electrical spare parts and special engineering material, store rooms for these purposes were fitted for the equipment intended to go into them. This was possible due to the delay in construction on the vessel on account of the War, the spares having been delivered so that boxes and parts could be measured and plans drawn for their proper stowage.

Compartment B-105 was set aside for all electrical and mechanical spare parts under the cognizance of the Bureau of Engineering, except the bulky spares which were stowed in compartment B-302 on account of the size of the access openings available. All electrical and mechanical spare parts under the cognizance of the Bureau of Construction and Repair and electrical spares under the cognizance of the Bureau of Ordnance were stowed in Compartment A-514. Compartment C-304 was reserved for the stowage of heavy lifting gear parts, and for items removed from machinery for safekeeping when visiting Navy Yards.

The engineer officer signed custody receipts for all material in these storerooms, supplied his own storeroom keepers, and took the entire responsibility for their accounting, issue and replacement. The allowance books were indexed with the exact location of each item.

## TRIALS.

The final official trials of the *Tennessee* were held from May 12 to 16, 1921. Just previous to the trials the ship was docked at Boston, Mass., for cleaning and painting the underbody.

The ship averaged 21 knots on the high speed runs at a displacement of 33,367 tons, developing an average shaft horsepower of 29,868. The maximum shaft horsepower developed was 30,909.

Below is given the principal average data obtained on the trial. Where propelling machinery is referred to it includes certain of the main turbine auxiliaries as follows: Exciter, main circulating pump, condensate pump, air ejectors, oil cooler circulating pump and lubricating oil pump. It also includes the main motor ventilating fans and liquid rheostat pumps.

BACKING TRIAL DATA.

| | From full power. | From 19 knots. | From full power one generator. | From 15 knots. |
|---|---|---|---|---|
| R.P.M. just before signal to back | 169.5 | 160.1 | 130.4 | 127.7 |
| Time of signal to back, minutes and seconds | 0 – 0 | 0 – 0 | 0 – 0 | 0 – 0 |
| Motors started to back, minutes and seconds | 0 –15 | 0 –17 | ... | ... |
| Ship dead in water, minutes and seconds | 3 – 0 | 2 –58 | 4 –16 | 3 –52 |
| Head reach, yards | 1009 | 730 | 915 | 840 |
| Average R.P.M. maximum backing | 130 | 130 | 100 | 100 |

All trials were successfully completed without machinery casualty of any kind. All contract guarantees were fully met.

## STEAMING TRIAL DATA.

| | 4 hour full power trial. | 8 hour 28000 S.H.P. trial. | 12 hour 19 knot trial. | 12 hour 15 knot trial. | 12 hour 10 knot trial. |
|---|---|---|---|---|---|
| Average speed, knots | 21.01 | 20.75 | 19.00 | 15.00 | 9.99 |
| Displacement, tons | 32,878 | 33,025 | 33,233 | 33,388 | 33,245 |
| Average, S. H. P. | 30,229 | 27,944 | 20,463 | 9,357 | 2,762 |
| Total oil consumption per hour (18750 B.T.U.) | 34,304 | 30,218 | 21,746 | 9,730 | 6,436 |
| Oil consumption per S.H.P. per hour | 1.135 | 1.081 | 1.063 | 1.040 | 2.33* |
| Oil consumed per square foot heating surface per hour | .821 | .723 | .521 | .466 | .411 |
| Oil consumption per knot pounds | 1633 | 1456 | 1144 | 649 | 644* |
| Water consumption, main propelling machinery per S.H.P. per hour | 12.486 | 11.976 | 11.568 | 11.582 | 14.904 |
| Number of boilers used during trial | 8 | 8 | 8 | 4 | 3 |
| Steam at boilers, pounds gage | 270 | 271 | 263.5 | 260.4 | 260.1 |
| turbine, pounds gage | 246 | 249 | 252 | 250 | 253 |
| turbine, temperature degrees F. | 419 | 427 | 432 | 427 | 430 |
| Vacuum, inches | 28.62 | 29.00 | 29.0 | 29.4 | 29.4 |
| Barometer, inches | ..... | ..... | ..... | ..... | 30.17 |
| Fireroom, air pressure, inches of water | 5.5 | 4.7 | 3. | 3.4 | 2.8 |
| Shaft, R.P.M. | 170.8 | 167.2 | 152.3 | 117.8 | 77.9 |
| Slip of propellers, per cent | 12.51 | 11.72 | ..... | ..... | ..... |

*On account of having to use No. 1 Nozzle partially open on the after turbine on the 12 hour 10 knot run, a short run of two hours was made at 10 knots with the forward turbine which showed oil consumption of 1.91 pounds per S.H.P. hour and 505 pounds of oil per knot.

Special reversing and turning tests were run in order to subject the machinery to severe tests and to obtain necessary design data.

# HYDRAULIC STEERING GEARS.

By Commander H. S. Howard (C. C.), Member.

1. In our latest battleships hydraulic steering gears have been installed. These steering gears represent a rather elaborate and complicated set of machinery a full understanding of which is necessary, or at the very least an understanding of the principles of which is necessary, in order that the full benefit of this gear may be derived, and successful and satisfactory operation assured. This is particularly true since the introduction or rather re-introduction of hydraulic steering gears in our Navy represents a distinct change from the well-established practice of many years.

2. Hydraulic steering gears are not a completely new installation in the United States Navy, for such steering gears, and in fact hydraulic machinery in general, was standard equipment on ships built by the Union Iron Works for many years. Thus the old *Monterey*, the *Olympia*, *Oregon*, *Wisconsin*, *Ohio*, *Cheyenne* and a number of other naval vessels were equipped with hydraulic steering gears. The principal reason for adopting hydraulic power on all but the latest of these ships seems to have been the advantage claimed therefor over steam power for operating auxiliaries, for at that time electricity was not in general use on shipboard and our Navy had not become in any sense an electrical one. It was claimed that if hydraulic power were used, steam pipes with their liability to serious damage in action, with consequent injury to the ship and personnel, need not be led outside the engine and boiler rooms; the elaborate care required for auxiliary steam engines would be unnecessary; hydraulic machinery would not

need warming up before running; and finally the hydraulic motors were much simpler than steam engines. These points of superiority are of rather academic interest now, but they are still of interest in considering this general subject.

3. The steering gears of all these older ships are practically the same. To the rudder head is attached a cross head from which connecting rods, divided into upper and lower branches, lead forward to wrist pins in blocks confined to a fore and aft motion by guides. To these blocks connect the piston rods of two hydraulic cylinders, the cylinders being located between the upper and lower branches of the rudder crosshead connecting rods. The pistons are double acting, the forward end of each cylinder being connected to the after end of the other cylinder. These pipes are connected to the control gear by pipes which in turn become supply or exhaust as desired, depending upon the direction in which the rudder is to be moved. The selection depends upon a so-called differential valve which is controlled from the steering position. Pressure from the pumps and accumulators is then supplied direct through the differential valve to the steering gear cylinders. A follow-up gear is provided which closes this valve after the steering wheel has been stopped, thus holding the rudder securely at the desired angle, since connection from the supply to the exhaust pipe, or vice versa, is closed off.

4. In general then, this older form of hydraulic steering gear consists of two hydraulic cylinders, supplied direct with pressure by pumps and accumulators, the pistons of these cylinders connecting through piston and connecting rods to the rudder head.

5. Notwithstanding their satisfactory operation, hydraulic steering gears were not generally adopted in the Navy, and steam steering engines continued to be the standard practice. However, with the general introduction of electricity as a source of auxiliary power in our ships, and the increasing familiarity of the operating personnel with electrical machinery,

it seemed desirable to apply electricity, with its obvious advantages, to the steering gear. Consequently, after experimental installations on five smaller vessels, electrical steering gears were installed as auxiliary to the steam gear on the battleships *New York* and *Texas*. The motors for this auxiliary gear are of 150 H.P. and operate the standard right and left handed screw gear in a manner identical to that used with the steam gear. The next step, having found success with electrical gear as an auxiliary, was to install this gear in power equal to the steam gear and this was done on the *Nevada*, two 150 H.P. motors being installed, these operating together to give 300 H.P. From here the natural consequence followed, and on the *Oklahoma*, *Pennsylvania*, *Arizona*, *Idaho* and *Mississippi*, the electrical steering gear became the main gear and the steam gear the reserve. On the *Arizona* a 300 H.P. motor is used while each of the other four vessels is equipped with a motor of 350 H.P.

6. However, experience with the electrical gears on the *New York* and *Texas* and the larger gears on the *Arizona* and other ships brought out a point of inefficiency in gears of this type. This was the necessity of starting and stopping the large motors each time the rudder was moved, which involved overcoming the inertia of the armature with consequent losses evidenced by heating of the motors. Further, the motors in these last ships were of great power, 350 H.P., and it was seen that if capital ships continued to increase in displacement and speed the steering motors would become almost prohibitive in size.

7. For a number of years hydraulic speed gears had been used for turret turning and experience had shown them to be most successful in operation. In these gears, of which the Waterbury Tool Company's type had been used in the Navy, a variable stroke pump of special type is used in connection with a constantly running motor. In turret turning, the great advantage of this variable stroke pump is that, depending upon the length of the stroke of the pistons in the pump, more or

less oil will be pumped and as a consequence the rate of training of the turret can be varied from a high speed to a very slow speed, in fact to zero speed, the changes in speed occurring without break. After some consideration of the subject it appeared that a pump of this type would for several reasons be inherently suitable for supplying power for the steering gear. First of all the motor would be continuously running, and running at practically no load when the rudder was not being operated. Second, when under load, the capacity of the pump could be so varied through regulating the stroke of the pistons, that large amounts of oil pumped and consequent rapid movements of the rudder could be combined with low loads, and small amounts of oil pumped and large torque combined with severe conditions of load on the rudder. This latter point is of the greatest importance, for there is probably no machinery on board ship which must meet more severe and rapidly changing conditions of load than the steering gear, and must at the same time possess absolute reliability.

8. These various considerations brought about a return to hydraulic steering gear for our capital ships. The fact that a direct push is desired to handle the rudder through the rudder stock brought a return to the hydraulic cylinders of the old *Olympia, Oregon* or other ships, but as far as the remainder of the gear is concerned, there is this important difference; in the older ships constant hydraulic pressure and flow of liquid was maintained to the motor cylinders, while in the new gear, the flow of oil varies with the conditions of load, the torque increasing with reduced flow of oil to meet high load, and decreasing with increased flow of oil on ordinary low loads. This great improvement as compared with the old gears was brought about by the introduction of the variable stroke pump.

9. It was decided to install a steering gear of this type on the *New Mexico*, and investigation showed that with this new type of gear it would be necessary to use only a motor less than ¼ the power of that installed on sister ships equipped with

straight electric steering gear. The advantages were at once apparent and they became so obvious on further study that this same type of gear was eventually adopted for all other capital ships building, and even for scout cruisers.

10. The *Tennessee* steering gear was developed directly from that installed on the *New Mexico*, and a description of the former gear will apply equally well to both. On the *Maryland* and *West Virginia* some differences were introduced, these differences being principally in details. Both the *Tennessee* and the *Maryland* have recently completed official trials and a description of their gears, based also upon the performance on trial, will give, it is believed, a general understanding not only of these gears but of other future gears now under construction.

11. A diagrammatic layout of the *Tennessee* gear is shown in figure 1. The rudder crosshead is connected by means of rods to two 20-inch diameter rams which work in 4 hydraulic cylinders located in the after compartment. From these cylinders two main pipes lead forward through two intervening compartments to the main pump sets. These main pump sets are in duplicate, each consisting of a shunt motor driving two Waterbury pumps. The pump lines connect through 3-way valves to the main lines to the ram cylinders, these 3-way valves permitting rapid shift from one main pump set to the other. Electrical and piping connections are such that but one main set can be operated at a time so that one set is always a stand-by for the other. The angle of the valve plates of the Waterbury pumps, which angle controls the amount of oil pumped and consequently the movement of the rudder, is controlled through gearing by small compound wound pilot motors which are in duplicate, as a stand-by measure, these pilot motors and gearing being also located in the main pump room. The pilot motors are controlled electrically from the bridge and other steering stations.

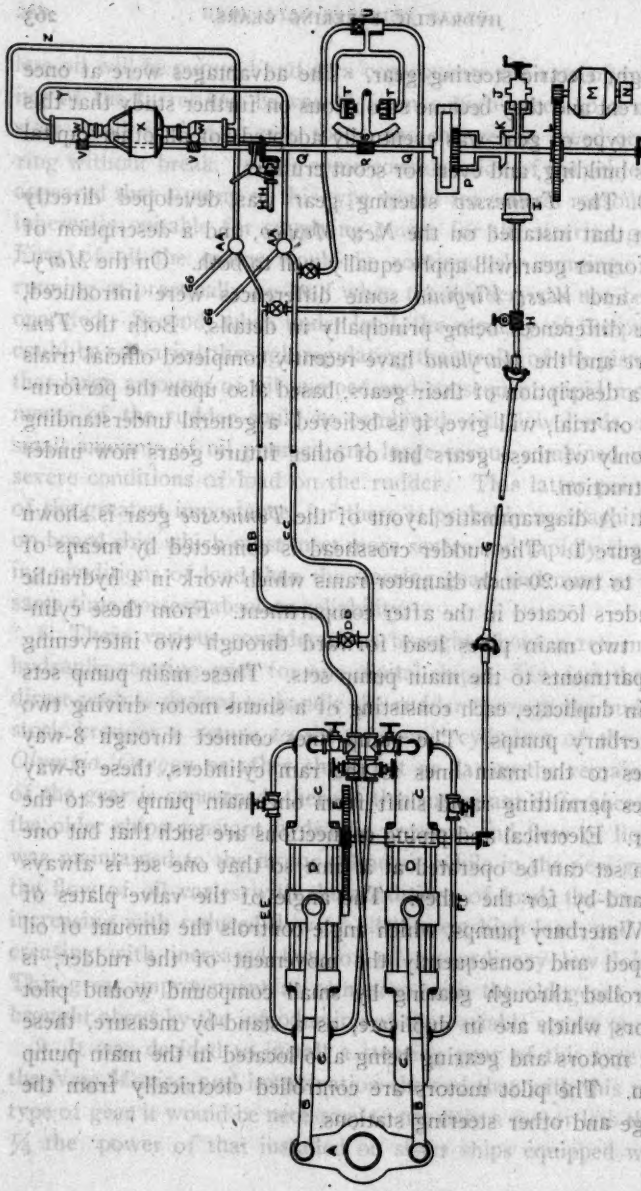


FIG. 1.—DIAGRAM OF "TENNESSEE" STEERING GEAR.

"TENNESSEE" STEERING GEAR.

| | |
|---|---|
| A | Rudder Crosshead. |
| B, B | Crosshead Connecting Rods. |
| C, C, C, C | Main Cylinders. |
| D, D | Rams. |
| E | Follow-up Gear Rack. |
| F | Bevel Gears; follow-up gear. |
| G | Follow-up Shaft. |
| H | Indicator and Limit Switch. |
| I | Slip Clutch. |
| J | Clutch. |
| K | Gears to pilot shaft; shown bevel, actually worm. |
| L | Spur Gear to Pilot Motor. |
| M | Pilot Motor. |
| N | Magnetic Brake. |
| O | Trick Wheel. |
| P | Differential Gear. |
| Q | Control Shaft. |
| R | Angle Gear Worm. |
| S | Angle Gear Quadrant. |
| T, T | Pressure Control Pistons and Stops. |
| U | Pressure Control Cylinder and Shuttle Valve. |
| V, V | Worm Gears controlling angle of Valve Plates. |
| W, W | Main Pumps of 1 Set. |
| X | Main Motor of 1 Set. |
| Y, Z | Pipe Lines from Main Pumps. |
| AA, AA | 3-Way Valves. |
| BB, CC | Main Pipe Leads to Cylinders. |
| DD | Relief Valve. |
| EE, FF | By-pass Pipes. |
| GG | Pipe Leads from Main Pump Set not shown. |
| HH | Clutch between Main Pump Sets. |

12. An essential part of this gear is the follow-up feature, for without a follow-up gear on the hydraulic part of the mechanism, the pumps would commence pumping oil when the valve plate was moved off neutral by the pilot motor, and would continue to pump oil until the rudder moved over to the extreme angle against the stops, even with the pilot motor stopped. To prevent such action a follow-up gear is introduced which consists of shafts and gearing leading from one ram to a differential gear in the pilot motor shaft. As soon as the rams and consequently the rudder commence to move, due to movement of the pilot motor shaft and angularity of the main pump valve plates, this follow-up shaft tends to reduce the angle of the valve plates, through the differential gear. When the pilot motor is stopped, this follow-up motion brings the valve plates to neutral, oil is no longer pumped and the whole gear comes to rest.

13. Another most important feature of the gear is the so-called torque equalizer, or pressure control. In describing the reasons for the adoption of hydraulic steering gears, one of the main advantages of the variable stroke pump for this work was stated to be its ability to increase torque to meet great loads by reducing piston stroke and thereby the amount of oil pumped, with consequent increased torque, due to keeping at the same time the full speed and power of the driving motor. To realize this advantage it becomes necessary to introduce some means of reducing the pump valve plate angle under heavy loads. The means taken to accomplish this desired result consists of limiting the movement of the valve plate control shaft. On the shaft is an angle worm gearing into a quadrant and the movement of the latter is limited by stops actuated by pistons. These pistons are within cylinders to which line pressure is admitted, the pistons being ordinarily held in the retracted position by springs. When the line pressure becomes sufficiently great to cause the pistons to move out against their retaining springs, the stops come into play

and limit the opening of the main valve plates. If the pressure becomes very high the valve plates will be forced back to neutral and no oil at all will be pumped. This feature of the steering gear is in effect an automatic control to meet varying load conditions.

14. The possibility of breakdown of the main gear, due to failure of electric power, motor or pump trouble, etc., has to be considered. To provide against disablement of the ship from this cause auxiliary steering gear is installed, the location being in a compartment just abaft the main pumps. This gear is also hydraulic and operates the main rams and rudder in a manner similar to that of the main gear.

15. A diagrammatic arrangement of the auxiliary gear on the *Tennessee* is shown in figure 2. This gear consists primarily of a motor for which the power is furnished by a storage battery. This motor drives a Waterbury pump, a duplicate of the main pumps, through two sets of spur gears, and the pipe lines from the pump are led aft to the main ram cylinders with connections thereto independent of those from the main pumps. In this way a completely independent unit is provided. In addition to the auxiliary electric motor the pump can also be operated by an air motor or by hand wheels, connection being made to any of the three motors by suitable clutches. For this auxiliary gear, the amount of oil pumped and consequently the rate of movement of the rudder is controlled by a hand wheel which directly controls the angle of the valve plate of the pump. There is no connection with the follow-up gear, so it is necessary for a clutch in the follow-up shaft to be disengaged when the auxiliary gear is in operation.

16. In both the main and auxiliary systems, in fact in any hydraulic system, some means must be provided to keep the system absolutely full of oil. Expansion of the oil, due to increased temperature, will cause an excess supply of oil, with consequent excessive pressures, if relief be not provided, while

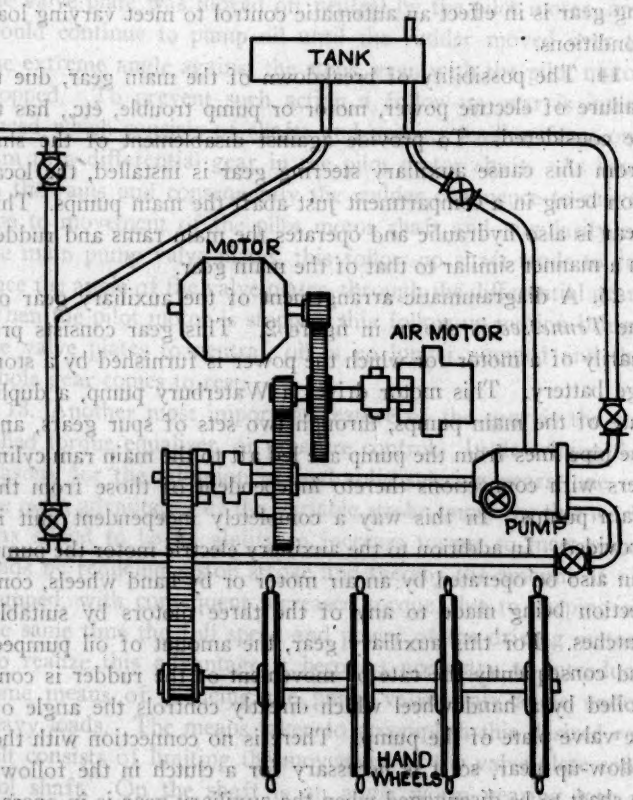



FIG. 2.—DIAGRAM OF "TENNESSEE" AUXILIARY STEERING GEAR.

contraction of the oil due to reduced temperature, or leakage, will reduce the amount of oil in the system; excessive pressures due to expansion cannot be allowed, while a hydraulic system in which air is present will not be satisfactory; the operation will be irregular, and noisy water hammers with possible damage to piping, fittings, etc., will result. To this end care is taken to lead all pipes so as to avoid air pockets and from the highest point of each system is taken a pipe connection to a replenishing tank with a vent at the top. This replenishing or gravity tank receives excess oil, or supplies additional oil as needed. Where the connections are made to the pipe lines a special shuttle valve of check type is fitted. The construction of the valve is such that connection is open to the low pressure line, thus allowing air to escape and oil to flow back to take its place, or relieving an excess pressure in this line. The connection is closed on the high pressure side, thus allowing no decrease in pressure on this side.

17. The system is intended for operation at pressures less than 600 pounds, this being the maximum working pressure to meet the designed conditions of operation at full speed of the ship.

18. However, during the operations of the ship and the steering gear, various conditions may arise which will cause the pressure in the ram cylinders to rise above safe limits. A possible cause of such an event is backing full speed with the rudder well over, or jamming the rudder against a dock in backing out from a berth. While the steering gear is designed and intended to control the movements of the ship under way, it is conceivable that from these or other causes the pressure in the system may rise to a point beyond safe limits. To meet this eventuality a by-pass relief valve is installed between the two main lines. Now as this relief valve lifts, the rams and the rudder will move due to a path being provided for the oil from one cylinder to another. The movement of the rams will naturally cause motion of the follow-up gear. This motion may

be sudden and to prevent this sudden motion wrecking the whole follow-up and pilot motor gear, a slip clutch is installed in the follow-up shaft. The setting of the relief valve and of the slip clutch should therefore bear a close relation to each other. This slip clutch may also come into play should the clutch to the follow-up not be disengaged when operating the auxiliary gear.

19. With the massive construction of the cylinders actual damage is a rather remote possibility, certainly without the whole gear being put out of commission, but leaky cylinder packing is conceivable and could not be allowed, due to the loss of oil in the system which would result. Some arrangement is therefore necessary for cutting out a leaky cylinder and this is provided for by by-pass valves in the main lines. In cutting out a cylinder it is necessary at the same time to cut out a corresponding cylinder working in the opposite direction, so as to balance the flow of oil from the pumps. To permit this operation the idle cylinders should be drained and connections are provided for this purpose. Under these circumstances it should be noted that the movement of the rudder will be at twice the ordinary speed, since the oil which is ordinarily delivered to two cylinders will be delivered to one. Care will have to be taken as a consequence in handling the rudder to avoid excessive loads on the steering gear. This precaution it should be possible to heed, as the cutting out of a cylinder should be a most unusual circumstance and should only be done until an opportunity occurs to undertake repairs to the cylinder out of commission.

20. To hold the rudder in central position at anchor, for overhauling the gear, or to permit steering by the propellers in the event that complete breakdown of the gear should take place, a centering pin is provided, worked from the ram room. This centering pin is simply a geared pin which can be lowered into position to hold the rudder amidships when desired, as for example at anchor or when the steering gear is to be over-

hauled. The pin must be raised before the steering gear is operated.

21. The only remaining features of the main gear are the limit switch and indicator. These are operated by a worm on the follow-up shaft. The indicator is installed to show the angle of the rudder, and the limit switch to stop the pilot motor when the rudder is at the limit of its travel.

22. The *Maryland* gear is in principle the same as that of the *Tennessee* or *New Mexico*, but in the arrangement of the control gear and lead of piping it represents a change.

23. The control system, including the follow-up, in the *Maryland* gear, consists, instead of rotating shafts and gears as on the *Tennessee*, of a push and pull system of rods, levers and bell cranks. This brings in other changes as direct corollaries.

24. The torque equalizing, or pressure control, gear is arranged for in this case by providing the lever, acted upon by the pilot motor to operate the valve plates of the main pumps, with a floating fulcrum. This floating fulcrum is balanced between a spring and a piston working within the torque equalizing cylinder to which the pressure of the system is admitted. The design is intended to be such that throughout the normal range of pressures no movement of this fulcrum will take place, while with excessive pressures, the pressure in the torque equalizing cylinder will overcome the spring, allowing the fulcrum to move and reduce thereby the angularity of the main pump valve plates.

25. The system, as stated, being push and pull, a slip clutch cannot be used in the follow-up rod as in the *Tennessee* gear as a protection, but instead the rod is actually broken by sections being omitted and heavy springs inserted instead. This prevents wreckage of the follow-up and control system in the event of the relief valve lifting or of the rudder taking charge.

26. The piping system consists of individual pipes to each cylinder and connection between these lines and the lines from the pumps is made by means of a manifold in the pump room instead of through 3-way valves. This makes it possible to arrange for almost any combination of working cylinders by means of opening the proper valves in this manifold in the pump room.

27. The auxiliary gear unit of the *Maryland* gear is practically the same as that of the *Tennessee*. It consists of a Waterbury pump, a duplicate of each of the main pumps, driven primarily by a motor which derives its power from a storage battery. Secondary sources of power are, as in the other case, an air motor or hand wheels. The pipe leads of this auxiliary gear, however, do not lead directly to the main cylinders, but connect direct into the lines from the main pumps, so that their control is through the manifold in the pump room to which reference was made above.

28. So far we have in service but four ships with hydraulic steering gears of the types described above; the *New Mexico*, *Tennessee*, *California* and *Maryland*, and the latter completed her trials but a short time ago. On the trials of the *New Mexico*, *Tennessee* and *Maryland* careful note was made of the operation of this gear and an attempt was made to obtain a full record of the oil pressures within the system, by means of graphic recording instruments, in order that a complete study and analysis might be made of the loads met with under various conditions of speed and helm. However, it was found that the recording instruments gave such unsatisfactory readings under conditions of sudden high pressures, that practically nothing could be deduced from the records. Changes were made in the instruments by introducing dampening coils but without success, the vibration of the pens under the sudden pressure loads giving records of no real value. However, certain events took place which gave valuable information, and these combined with reports of service experience suggest

changes which will, it is believed, improve the operation of these gears in the future.

29. The principal feature of the gear which apparently requires modification is the pressure control or torque equalizing apparatus. The principal function of this gear is as previously stated to act as a regulator, to enable the steering gear as a whole to meet heavy loads. These loads in general were found to come with great suddenness, this being shown by the action of the pressure-recording apparatus, and while spring balanced, pressure operated pistons might well act as intended with a gradually increasing, or changing pressure, it could not be seen that they came into operation under the actual conditions of sudden high pressure. Moreover, it was found that by improper location of the point of action of the pressure control gear, it was possible for this gear to act in a way quite opposite to that intended. Conditions might arise in which, with the whole gear at rest and valve plates in neutral, with no movement intended from the steering station, the increasing pressure due to the ship gathering speed ahead or astern might, through the pressure control system, actually bring about a movement of the valve plates which would either overload the motors, if the pressure were very high, or might even cause the pumps to operate in a direction which would allow the rudder to take charge.

30. These are rather remote possibilities, or at least it may be said that, while they are theoretically possible at any time, the actual conditions of operation of the ship and steering gear are such that it might be that unpleasant consequences would not arise within a life time. However, under severe conditions these possibilities are present and it is necessary that as far as possible, every part of the steering gear must fulfill its duty to the utmost. Where such conditions arise therefore, changes must be made and where these have been found necessary, they are, it may be noted, already under way.

31. A study of this subject has led to a completely different solution for equalizing the torque or regulating the load applicable to any gear of this type. This scheme seems to promise success. It is proposed to deal with the load taken by the main pump motor, as expressed in ampères, instead of dealing with pressure in the system. The load on the main motor is a direct measure of the pressure in the system and in general, this ampèrage, which denotes the load, is not subject to the violent fluctuations of pressure which were found to exist in the oil pressure. Now, if with increasing ampèrage, relays operate to introduce resistance into the pilot motor circuit, the pilot motor will slow down, reducing the angle of the main pump valve plates, causing less oil to be pumped and increasing thereby the torque which is the object sought. It may be argued that relief given in this way will be too slow in action, since pressure must rise to increase the load on the main motor, the latter must slow down to build up back E.M.F. and thereby increase ampèrage, after which the resistance must be thrown into the pilot motor circuit and this motor slow down. On the other hand, with modern efficient motors all these operations will take place practically instantaneously. Moreover, this system has the great advantage of meeting any condition of high pressure or load; whether the ship be going ahead or backing, increase in pressure always operates to increase torque, and with the gear at rest no load, no matter how great, can open the main pump valve plates.

32. Failures of hydraulic steering gears under test or operation seem to have been thus far, due almost entirely to the electrical features. Of the electrical causes the principal one is loss of power on the ship's circuit in use; another is stoppage of the motor of the main pumps, due to a heavy sudden overload, caused, for example, by an attempt to throw the rudder rapidly from hard over to hard over with the ship going full speed ahead. A third particular failure occurred on the *Tennessee* trials, with the ship going full speed astern. The

*Tennessee* differs from the *New Mexico* and *Maryland* in that it is possible to obtain a higher speed astern, which, of course, makes it possible to bring unusual pressures on the rudder, which is balanced for ahead, but not for astern motion. In this case the ship was backing over the measured mile at full speed, making in the neighborhood of 17 knots. The rudder was put over about 10 degrees, when it suddenly took charge and went over against the hard over stop. The explanation which was deduced afterwards, for at the time things happened too quickly for observation, was that with the very high pressure brought about by the conditions described, the magnetic clutch on the pilot motor was not sufficiently strong to hold this part of the gear as a fulcrum, the follow-up gear opened the valve plates of the main pumps and the pumps then operated to allow the rudder to go hard over. The case in question was a most unusual one and stronger magnetic clutches are being installed to avoid a similar happening in the future, although it is, of course, a rather remote possibility that the necessity should ever arise of running a battleship 17 knots astern. To return, then, to the two principal causes; *i. e.*, failure of power, or stoppage of motor due to overload. Both are evidenced in the same way, by the main pump motor stopping, and such an occurrence is extremely undesirable, for even though the auxiliary set be started at once (on the *Maryland* the auxiliary set is regularly started up and takes charge of the rudder in 10 seconds) control of the ship is temporarily lost from the steering station. Moreover, with such stoppage, the chances are great that the valve plates of the pumps will be at an angle. To start up the motor again it must be started under load which is almost out of the question with shunt motors, or the valve plates must be brought to neutral. To meet these conditions, it is now proposed that first of all such pilot lights or other visual or audible signals be provided that it will be known at once when power on a circuit has failed, so that the steering gear crew can switch over the set to a live

source of power. In addition, by the substitution of compound wound motors, for the present shunt wound motors, capable of starting under full load, combined with automatic resetting relays, a main pump motor would, if stopped by failure of power, or overload conditions, restart itself as soon as switched to a live source of power, or as soon as the load on the system had been somewhat reduced due to the swinging of the ship. The fact that in all existing ships duplicate main pump sets are provided, supplied with three sources of power, together with auxiliary sets to bridge over the time between the stopping of one main set from any cause and the starting of another, actually gives them good protection, but the changes outlined should, it is believed, result in improving the gears very greatly along the lines of automatic operation.

33. It has been brought out that to start a large compound motor under load as just suggested would impose a heavy and undesirable peak load on the ship's generators. As counter suggestions, therefore, it has been proposed that an automatic switch be developed which would shift the working motor from a dead source of power to a live source, or that an automatic device be developed which will if a main pump motor stop due to overload, with the pump valve plates at an angle, return these valve plates to neutral. The motor, if provided with an automatic resetting relay, would at once start up. These proposed devices would obviate the necessity for the introduction of compound-wound motors and would therefore represent a distinct improvement in the gear. Pending their development, however, compound motors seem to be almost essential for reliability of the gear.

34. The discussion of sources of failure in the preceding paragraph leads naturally to a consideration of auxiliary gear. An auxiliary gear is an absolute necessity, even with the duplicate main pumping sets provided, for first of all an auxiliary gear is necessary, at least with present arrangement of steering gear, to carry over the control of the ship while one set is

stopped due to failure from some cause, and before the second set is started up; and secondly, this gear is needed to operate the steering gear in the event that the leads furnishing power be damaged. The deduction from this, first of all, is that the most important feature of the auxiliary gear is a source of power completely independent of the source supplying the main pump set; this leads to the requirement for a local storage battery. However, the question then arises as to whether this storage battery should be of sufficient size to drive a main pump set motor, or simply to drive an auxiliary motor and pump as in the present arrangement.

35. The first arrangement, *i. e.*, a storage battery of sufficient power to drive one of the main motors seems to possess many advantages; it is available for either main motor; it may be floated on the line with an automatic switch which will keep the battery on charge while the source of power to the main motors is available, but will at once throw the battery to discharge if power fails; and since the control will not change, no indication will be had at the steering station that the auxiliary gear has stepped in to relieve failure of a main source of power. However, this storage battery will be considerably heavier than the present battery, auxiliary motor and auxiliary pump. Moreover, it is believed that with proper indications for loss of power, possible automatic switching from one source of power to another, together with compound wound main pump motors and automatic resetting relays, the necessity for an auxiliary gear carrying over the period between the stopping of one main set and the starting of another will be very slight. This leaves the auxiliary set as a distinct last resort means of steering, in the event of breakdown of power, main pump set motors, or any other feature of the main gears. To meet these conditions, the combination of auxiliary storage battery, motor and pump seems to promise the best solution. It is not out of place here to speak of hand gear, or any other type of gear in combination with the

auxiliary set. This seems entirely unneeded, particularly as a large modern ship could probably not be steered satisfactorily by the hand gear, while the latter has the disadvantage of taking up a great deal of space. The hand wheels will probably therefore be omitted on future ships.

36. About the only essential remaining point of difference between the two gears described, and a point which is worth discussing as affecting the operation of such steering gears, and installations in the future, is the arrangement and lead of piping between the pumps and the main cylinders. It will be recalled that in the case of the *Tennessee*, the pipe leads from the main pumps are connected to three-way valves, to permit a quick shift from one main set to the other, and from these valves two main pipes lead to the ram cylinders. Further, the auxiliary piping leads independently to the ram cylinders. On the other hand, in the *Maryland*, the pipe leads from the pumps connect to manifolds from which four main pipes lead aft, one to each ram cylinder. Also the auxiliary pump piping connects into the main pump manifolds. The four-pipe arrangement on the *Maryland* seems decidedly preferable, for by this arrangement it is possible to operate the gear on various combinations of cylinders in the event of leaky packing in one cylinder, and moreover, the fact that four pipe leads are provided does give somewhat greater security against possible damage. In the *Tennessee* arrangement, to permit the gear to be run on two cylinders, by-passes in the ram room at the main cylinders themselves would have to be opened, and in the remote event of damage to one main pipe, or heavy leakage at a flange, etc., the gear could not be operated at all. Moreover, in the *Tennessee*, the separate auxiliary pipes in themselves present a rather remote source of trouble, for a leak in an auxiliary pipe, even with this gear idle, might prevent the whole steering gear from operating. The arrangement, then, which seems to promise the most satisfactory operation for the future, would be a system of piping as in

the *Maryland, i. e.,* an independent pipe to each cylinder, combined with three-way valves as in the *Tennessee* and stop valves in each pipe, together with a system of auxiliary piping as in the *Maryland.*

37. The advantages of hydraulic steering gears for large, high speed naval vessels are such that they undoubtedly have come to stay, and the more they are studied, the greater these advantages appear as compared with other types of steering gears installed in our ships. However, as stated at the very beginning of this article, it is necessary that the gear should be fully understood, to realize these advantages, or in fact to avoid possible troubles. The description and discussion which have been given are only general, but it is believed they are sufficient to give an understanding of the general principles and operation. If, then, by these notes, information which will be of value to Engineer Officers throughout the Navy in making them familiar with gears of this type has been given, and if general thought and discussion by engineers, marine or electrical, on the subject may follow, with the consequent improvement in our steering gears, it will be felt that the object sought for has been accomplished.

# DIRECTIONAL STABILITY OF AUTOMATICALLY STEERED BODIES.

By N. Minorsky.

## 1. INTRODUCTION.

The problem of directional stability of automatically steered ships is gradually becoming of increasingly greater importance for various reasons.

The possibility of obtaining more accurate steering by automatic means than can be accomplished by manual control with its inherent limitations due to the low sensitiveness of the human eye in detecting slow angular motions, fatigue, etc., becomes of greater importance with the increase in size of ships and cost of fuel.

For merchant ships an accurate and reliable automatic steering device becomes a real money saving proposition, largely justifying its use.

On battleships, by its use the absence or reduction of yawing in action means a better efficiency in gunfire, increased maneuvering speed and also a greater cruising radius.

In the case of airships, especially for long distance trips, directional stability is also of importance because the behavior of direction indicating instruments is then more satisfactory which leads to a still better stabilization, so that the certainty and safety of aeronavigation is thereby increased to an additionally greater degree.

A considerable number of other applications of the problem might be cited, such, for instance, as the radio control of ships and torpedos where the certainty and the accuracy of steered course must be assured between the moments when the radio signals establish a new setting of the controlling means.

Furthermore, quite recently, piloting vessels in harbors and channels by means of so-called radio cables carrying alternating current met with considerable success in this country and abroad*; it is likely that these installations will gradually replace older methods based on optical and acoustical signals. In that case an accurate and reliable automatic steering device enabling a vessel to follow very closely her "radio track" may be of considerable importance.

We may take a step further and predict some probable developments in the near future, which are already available, as far as principle is concerned and the existing handicaps are merely a matter of technical details.

Remarkable progress of recent years in connection with directive reception by coil antennas gives reason to believe that radiogoniometric methods of piloting ships or airships across still longer distances, such as trans-atlantic trips, before long will become an accomplished fact.

The main obstacle to it at the present time consists in the absence of a sufficiently powerful source of high frequency radiation adequate for radio compass performance.† There can be no doubt, however, that in the near future this obstacle will be overcome, especially in view of the continued progress along the line of high frequency generation by means of vacuum tubes.

The practical solution of the problem apparently will be reached when those technical difficulties are finally eliminated and when besides the magnetic and the true meridians given by magnetic or gyroscopic compasses respectively, the radio meridians or radio-routes will be established on the earth's surface.‡

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* A. Crossley, JOURNAL OF A. S. N. E., Feb., 1921. L. A. Herdt and R. W. Owens, Comptes Rendus, Jan. 3, 1921.

† In this connection the work of Lt. Cr. A. Hoyt Taylor, U. S. N. R. P., can be consulted; Bureau of Standards No. 353; the same subject is treated in Comptes Rendus, Jan. 3, 1921.

‡ Such attempts have already been made in Germany during the war. See Jahrbuch der Drahtl. Telegr., Feb., 1920.

We may therefore state that even at the present time we have various means to ascertain a fixed direction on the surface of earth and to utilize it accordingly either for the purpose of manual steering, or generally speaking, for the automatic steering, if we connect the rudder with the direction indicating apparatus whatever its nature may be (magnetic, gyroscopic or radio compass) by any appropriate means, for example by means of an ordinary follow up system.

The question therefore arises whether such solution is correct? We know from past experience with automobile torpedoes that such solution is quite possible. Experience, however, shows that this simple method is sufficient only when the mass, or better the moment of inertia of the steered body, is not great and when very accurate steering is not the main object. The difficulties of such positional control increase very rapidly with the increased size of the vessel. It is also known that no good steersman will steer the ship by observing only the compass card without taking into consideration the state of angular velocity of the vessel; such a steering would be very inefficient in so far as the effect (action of the rudder) would come always too late after the cause (the original disturbance) which contributes to the permanent yawing.

An efficient helmsman keeps the ship accurately on her course by exerting a properly timed "meeting" and "easing" action on the rudder, i.e., by taking into consideration the elements characterizing the motion from the dynamical standpoint, namely, the instantaneous angular velocity of yawing as well as its time variations.

It has often been stated that the human intuition of the helmsman cannot be replaced by any mechanical contrivance whatever its nature may be. Such a standpoint seems to be erroneous, as far as the problem of automatic steering is concerned, since there is not so much question of intuition as of suitable timing based on

actual observation. Once the element of observation is removed from the helmsman, there can be no accurate steering whatever his intuition may be.

The question, therefore, arises as to how the observation must be co-related to the timing of the rudder in order to obtain accurate steering?

If, therefore, accurate steering is nothing more than a special kind of timing of the rudder complicated by the inertia of the body to be steered, we may expect to be able to establish analytically what kind of timing must be adopted in order to reach the best possible conditions for directional stability of the body to be steered on its course.

An attempt to establish such fundamental relations for automatic steering is contained in the following.

## 2. LIMITATIONS UNDER WHICH THE PROBLEM CAN BE INVESTIGATED ANALYTICALLY.

The problem of automatic steering of moving bodies cannot be handled mathematically for the case of unlimited angular motion since there is no analytical expression applicable to the various torques acting on a ship, in general. It is practically necessary, therefore, to limit this investigation to the case where the mathematical solution has a real physical meaning. For this reason the stability of angular motion has been considered only for small deviations of the steered body from the desired direction which not only simplifies considerably the analytical solution of the problem but gives the only solution that is of practical interest, since only small deviations are possible, if the steering is to be accurate; if there is no stability for small deviations, it means that there is no stability in general since the ship will be continuously deviated from her course.*

* This method of attack is generally known as *method of small oscillations*. For more information see Webster, Dynamics, p. 157.

We may assume, therefore, that the angle of deviation of the ship from her course $\alpha$, the angular velocity $\frac{d\alpha}{dt}$ and the angular acceleration $\frac{d^2\alpha}{dt^2}$ of yawing are very small, belonging to the first order of small quantities with which we are now concerned, and that the higher orders may be neglected.

The magnitude of the first order quantities may be fixed arbitrarily by assigning to them the small values that are determined by practical considerations.

For example, it may be assumed that the angle of yawing must not exceed $\frac{1}{4}°$ (.0043 radians), and that the angular velocity of yawing is $3.5 \times 10^{-3}$ radians/sec., which values may be considered as limiting from the standpoint of their practical detection by the eye;* the square of those values being of the second order may be neglected. Having thus established the first order for the angles and velocities, the order of magnitude for angular accelerations can easily be ascertained, and the range within which the problem is studied is thus completely determined.

To fix ideas by a concrete example, the angular motion of a seagoing surface ship will be studied, although the method is general and can be applied to other cases.

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* The above figure $3.5 \times 10^{-3}$ rad./sec. for angular velocity results from a series of observations carried out by the writer in 1916 on the sensitiveness of the eye in detecting angular velocities during a short interval of time (2–5 seconds). These observations were made by placing helmsmen on a revolving table, the angular velocity of which was known to the writer both in magnitude and direction, but was unknown to the observer; a very good line of bearing was available and absence of rolling and pitching made the observations very easy. Nevertheless, no certainty was observed in detecting either the rotation or its direction (being given 2–5 sec. for observation), when the table was rotating slower than $3.5 \times 10^{-3} - 4.5 \times 10^{-3}$ radians/sec., which corresponds to 30–40 minutes for one complete revolution of the table. It may be mentioned that during the time these tests were made a gyroscopic angular velocity indicator, designed by the writer, was found to detect the velocity corresponding to the vertical component of the earth's rotation (latitude 60°) i.e. a rotation about 50 times smaller in rate than the angular velocity of the table above referred to; the angular momentum of the gyroscope used in those tests was about $2.8 \times 10^6$ gr$-$cm.$^2-$sec.$^{-1}$, the gyroscopic precessional torque by means of which the earth's rotation was manifested being about 170 gr. cm.

Assuming that a ship is to steer a course $\alpha_0$, the angular motion to be controlled by the rudder depends upon the single variable, or coordinate angle $\alpha$ giving the departure of the ship from her true course $\alpha_0$, as well as upon its time derivatives.

The other coordinates, defining completely the angular position of the steered body at any instant, namely the angles of roll and pitch, as well as their time derivatives, are not considered here since they are not controllable by the rudder.

Since the conditions of directional stability do not depend upon any particular value of the course $\alpha_0$ we may assume, for the sake of simplicity that $\alpha_0 = 0$ and consider only the deviations $\alpha$ from the true course due to yawing.

### 3. GENERAL FEATURES OF ANGULAR MOTION, VARIOUS TORQUES ACTING ON A SHIP.

In analyzing the turning motion of ships, Sir P. Watt says:

"The turning motion of a ship may be divided into three stages:

(*a*) When the rudder is first put over and the pressures on the hull are those necessary to produce angular acceleration.

(*b*) When accelerational forces are combined with those caused by the resistance of the ship to rotation, and

(*c*) When finally turning uniformly in a circular path.

The character of the forces acting during the states (*a*) and (*c*) can be ascertained and the type of motion under the complex conditions represented by (*b*) will consist of a gradual replacement of the motion at (*a*) by that at (*c*)."

It is apparent that for the purpose in view only the (*a*) stage of the motion is of interest.

We may enumerate now the torques acting on a ship and causing the initial turning motion as follows:

1. External disturbing torque due to any cause whatever (wind, waves, propellers, etc.)
2. Restoring torque due to rudder.
3. Resistance of the ship to turning.

There are no exact data relative to the nature and magnitude of the resistance of a ship to turning. Various authorities believe, however, that this resistance follows laws similar to those for rolling resistance, and that, probably, it may be expressed with practical accuracy by some law of proportionality to a certain power of angular velocity of yawing; there is no exact agreement, however, among investigators as to the exact value of the exponent, but it is generally agreed that it is somewhere between 1 and 2, varying according to a complex law. But, while the exact law, giving the resistance to turning as a function of angular velocity within a wide range of variations of angular velocity, remains unknown, we may safely assume that this law is linear within the narrow range of such variations around the zero velocity point.

Analytically this results from the well known method of approximation by infinitesimal analysis, according to which a small arc of continuous curve may be replaced by its chord without committing an error greater than the second order of small quantities, which therefore may be neglected.

The resisting torque, therefore, may be represented by: $-B\frac{d\alpha}{dt}$ where B is constant and the minus sign expresses the fact that the resisting torque always acts *against* the direction of angular velocity.

The other two torques acting on a ship are:

$C(\rho)$ the restoring torque due to the action of the rudder, which is in general, a certain known function of the angle of rudder $(\rho)$.

D, the disturbing torque deviating the ship from her course which may be due to any cause whatsoever as wind, waves, propellers, etc.; in general, this torque is so irregular that it cannot be taken into account analytically. It will be shown, however, that stability conditions do not require any mathematical expression for this torque.

Applying the principle of inertia in angular motion we have the expression $A\frac{d^2\alpha}{dt^2}$ equals the sum of all the combined external torques acting on the ship; A being the *effective* moment of inertia of the ship about the vertical axis passing through its center of gravity; it is generally greater than the moment of inertia of the ship on account of the inertia of water displaced by yawing, therefore,

$$A\frac{d^2\alpha}{dt^2} = -B\frac{d\alpha}{dt} - C(\rho) + D$$

The reason for the minus sign for $-B\frac{d\alpha}{dt}$ has been given above. It will be shown later that analogously $C(\rho)$ must also have the minus sign.

We may, therefore, write:

$$A\frac{d^2\alpha}{dt^2} + B\frac{d\alpha}{dt} + C(\rho) = D$$

Since the function $C(\rho)$ is always a continuous function of the angle $\rho$ because an infinitely small increase in the restoring torque corresponds to an infinitely small increase of this angle and the angle $\rho$ varies during accurate steering very little from its midship position ($\rho = 0$), we may expand the function $C(\rho)$ in Taylor's series around the point $\rho = 0$ and thus write:

$$C(\rho) = C(o) + \rho\left(\frac{dC}{d\rho}\right)_{\rho=0} + \frac{\rho^2}{2'}\left(\frac{d^2C}{d\rho^2}\right)_{\rho=0} + \frac{\rho^3}{3'}\left(\frac{d^3C}{d\rho^3}\right)_{\rho=0} + \ldots ( )$$

When the angle of rudder is zero obviously there is no restoring torque, thus $C(o) = o$. Further, $\rho$ being a very small angle the terms containing higher powers of $\rho$ may be neglected; we have then approximately

$$C(\rho) = \left(\frac{dC}{d\rho}\right)_{\rho=o} \times \rho$$

where

$$\left(\frac{dC}{d\rho}\right)_{\rho=o} = k$$

is a constant quantity characterizing the area of the rudder, lever arm of the rudder pressure, etc.

This gives a considerable simplification of the problem because under conditions that are of practical interest we may assume with sufficient accuracy that the restoring torque of the rudder is simply proportional to its angular position, the constant of proportionality $k$ being equal to

$$\left(\frac{dC}{d\rho}\right)_{\rho=o},$$

that is, to the value of the derivative of the rudder torque in respect to the angle of rudder for $\rho = o$.

The character of the function $C(\rho)$ is generally known; thus according to Joessel*

$$C(\rho) = P \cdot L = 4.6\, K\, A'\, V^2\, L \frac{\operatorname{Sin} \rho}{G + H \operatorname{Sin} \rho},$$

where

P is the total pressure on the rudder in lbs.

V is the speed of ship in knots.

A′ is the area of the rudder in square feet.

K is a constant (for battleships and cruisers K varies between .6 and .75).

* Hibbard, Transactions A. I. E. E., 1914, p. 634.

L is lever arm of the rudder pressure equal to $l+d$, where $l$ is the distance from the forward edge of the rudder to the center of gravity and $d$ the distance of the center of pressure from the forward edge of the rudder.

G and H being two empirical constants; G=.39 and H=.61. This gives

$$k=\left(\frac{dC}{d\rho}\right)_{\rho=0}=\frac{4.6\ K\cdot A'\ V^2\ L}{G}.$$

## 4. THE PROBLEM OF AUTOMATIC STEERING.

The equation assumes then the form:

$$A\frac{d^2\alpha}{dt^2}+B\frac{d\alpha}{dt}+k\ \rho=D \qquad (2)$$

The problem is completely determined if the relation of the angle of rudder $\rho$ as a function of $\alpha$ and its time-derivatives is given.

The equation (2) is the basis for the various methods of automatic steering.

In the following only those devices that give a continuous regulation to the rudder will be considered because only such devices are able to follow closely the development of angular motion of the ship which is always continuous with time. An adequate reaction on the part of the rudder in such a case continuously follows the action of disturbance and keeps it in check at all times.

It is also only in the case of a continuous regulation that the problem can be attacked by analysis.

Several methods of regulation are possible; we may mention about two fundamental classes according to whether the regulation affects the angle of rudder or the rate of change of this angle.

Theoretically one could go still further and imagine the third class of devices in which the controlling means would act upon angular acceleration of the rudder, but this is of no practical interest as will be shown later.

The above fundamental methods of rudder control can be expressed by simple linear relations of the form:

$$\rho = m\,\alpha + n\,\frac{d\alpha}{dt} + p\,\frac{d^2\alpha}{dt^2} \quad \text{(x)} \qquad \text{First class.}$$

$$\frac{d\rho}{dt} = m_1\alpha + n_1\,\frac{d\alpha}{dt} + p_1\,\frac{d^2\alpha}{dt^2} \quad \text{(xx)} \qquad \text{Second class.}$$

$$\frac{d^2\rho}{dt^2} = m_2\alpha + n_2\,\frac{d\alpha}{dt} + p_2\,\frac{d^2\alpha}{dt} \quad \text{(xxx)} \qquad \text{Third class and so on.}$$

For the sake of better understanding the individual features of each component control derived from various controlling means, several particular cases are treated in the beginning. Thus in studying the first case, we will first assume that $n=0$, $p=0$ which brings the fundamental relation of rudder control to a simple form $\rho = m\cdot\alpha$; then we will consider the other two sub-classes of the first class namely, the cases when $m=0$, $p=0$ and when $m=0$, $n=0$ in the above general relation (x) characterizing the first class of rudder control.

This study is contained in the following section 5 under the headings (*a*) (*b*) (*c*) and the general cases corresponding to the equations (x) (xx) are treated in sections 6 and 7. It is apparent that the number of possible combinations is very considerable; no attempt is made, therefore, to treat them all. The general method of attacking the problem shown in the following permits, however, to extend easily the analysis to any particular combination.

The practical method of combining the above mentioned controlling means can be made of course in several ways and it is not intended to go into consideration of them all; incidentally it may be mentioned that a mechanical differential gear affords perhaps one of the simplest solutions

of such a linear combination; thus, for instance, if one component gear of the differential is turned over an angle proportional to $m\,\alpha$, and the other

$$n\frac{d\alpha}{dt}$$

the planetary wheel is turned over an angle proportional to

$$\frac{1}{2}\left(m\,\alpha+n\frac{d\alpha}{dt}\right),$$

and thus will afford the requisite linear combination. By adding through a second differential gear an angle proportional to

$$p\frac{d^2\alpha}{dt^2}$$

one will have on the planetary shaft of this second differential the complete linear relation

$$m\,\alpha+n\frac{d\alpha}{dt}+p\frac{d^2\alpha}{dt^2}$$

and can utilize it accordingly.

It is obvious that the above mentioned linear combination of controlling means is algebraic, i.e., holds not only for the magnitude of

$$m\,\alpha,\ n\frac{d\alpha}{dt},\ p\frac{d^2\alpha}{dt^2}$$

but also for their respective signs.

### 5. DEVICES OF THE FIRST CLASS (CONTROL OF THE ANGLE OF RUDDER).

As stated above, the controlling means act directly upon the angle of rudder; several methods of regulation are possible; they are successively treated in the following.

(a) *Positional control of the angle of rudder.*

This case is the simplest and oldest in the art; almost all inventions dealing with the problem of automatic steering are based on this principle which may be described as follows.

As soon as the ship deviates from her course over an angle a suitable mechanism moves the rudder over an angle $\rho$ proportional to the angle of deviation (or yawing) $\alpha$.

Analytically one may express this case by the very simple relation $\rho = \pm m\,\alpha$, where $m$ is a coefficient of proportionality, the double sign meaning that, in general, there are two ways of connecting the azimuth control to the rudder.

Substituting $\rho = \pm m\,\alpha$ into the equation (2) and putting $km = C$ we have:

$$A\frac{d^2\alpha}{dt^2} + B\frac{d\alpha}{dt} \pm C\alpha = D \qquad (3)$$

This is the well known equation for the pendulum; the motion is given by the relation:

$$\alpha = Me^{x_1 t} + Ne^{x_2 t},$$

$x_1$ and $x_2$ being the roots of the auxiliary equation $Ax^2 + Bx \pm C = 0$ (3′) namely:

$$x = \frac{B}{2A} \pm \sqrt{\frac{B^2 - 4AC}{4A^2}}$$

It is known that in order to have stability of motion the auxiliary equation (3′) must have no positive real roots which means that only the plus sign must be used before C; from the practical standpoint the above analytical condition simply means that the positional (otherwise: azimuth) control must be connected to the rudder in such a way that whenever a deviation $\alpha$ occurs a mechanism controlled by that deviation must so act as to oppose its further increase which thus constitutes a correct connection.

For analogous reasons the minus sign signifies that the controlling mechanism is connected with the rudder in the wrong way.

Depending upon whether $B^2-4\,AC$ is positive or negative there will be either exponential decay of yawing or damped oscillatory motion.

Denoting—

$$\frac{B}{2\,A}$$

by $u$ and the positive value of the root

$$\sqrt{\frac{B^2-4\,AC}{4\,A^2}}$$

by $v$, we have in the first case the solution of the form: $\alpha = Me^{-(u+v)t} + Ne^{-(u-v)t}$; and in the second the solution of the form: $Me^{-ut}\,\mathrm{Cos}\,(vt+s)$, where M, N, $s$ are constants of integration determined by boundary conditions. For a given ship B and A may be regarded as constant. C is a parameter characterizing the degree of the rudder action for a given value of deviation $\alpha$; for example, the larger the area of the rudder, the greater will be the constant C; likewise, for the same rudder, the greater the angle $\rho$ for a given angle of deviation $\alpha$—the greater C will be. Thus for a gradually increasing value of C a "dead beat" steering may become oscillatory and vice versa.

If C is very small (i.e. a very weak rudder action) $u$ is nearly equal to $v$ so that $u-v=0$ which means that it will take a long time to bring the ship back to her course, once she has deviated therefrom by some reason; this is also obvious for the reason that it was assumed that the action of the rudder is small. By gradually increasing C the stability of the course will be increased but the steering becomes oscillatory or nearly so.

The damping out of oscillations caused by a disturbance depends upon

$$u = \frac{B}{2A};$$

for gradually increasing size of ships the moment of inertia increases apparently more rapidly than the friction to turning B. Sir Philip Watt considers that in the initial phase of yawing the value of B is very small.

This, probably, constitutes the main reason why positional control, although being much used in steering smaller craft, such as automobile torpedoes, and the like, has been found inadequate for steering larger vessels; this also can be understood from the fact that those devices are incapable of performing any "easing" or "meeting" action attributed generally to the helmsman's intellect.

From the above study it is apparent that the real cause for such a deficiency is in the fact that the conditions of damping out the yawing are not obtained by a purely positional rudder control.

(*b*) *Angular velocity control of angle of rudder.*

Another method of rudder control is obtained when the angle of rudder is varied in proportion to the value of instantaneous angular velocity of yawing recorded for example by a gyroscopic angular velocity indicator.* This case, consequently is characterized by a relation between the angle of rudder $\rho$ and angular velocity, of the form:

$$\rho = \pm n \frac{d\alpha}{dt},$$

where $n$ again stands for a suitable coefficient of proportionality expressing the degree of coupling between the angle of rudder and the angular velocity of yawing. In-

* Those devices called gyro-turn indicators or gyrometers are based on the principle of measuring instantaneous angular velocity by registering precessional reactions of a gyroscope constrained to move with the vessel. A device of that nature, compensated for the rolling and pitching disturbances, is described in U. S. Patent 1,372,184.

troducing this expression in the fundamental equation of automatic steering, and omitting the disturbing torque $D^*$ we have:

$$A\frac{d^2\alpha}{dt^2}+(B+E)\frac{d\alpha}{dt}=0$$

where $E=nk$ is the term relating to intensity of rudder action, depending upon area of the rudder, as well as upon the coupling or between the angle of rudder $\rho$ and the angular velocity of the vessel.

The auxiliary equation corresponding to the differential equation of motion is of the form: $Ax^2+(B+E)x=0$ whose roots are:

$$x=0;\ x=-\frac{B+E}{A}.$$

The motion is consequently of the form:

$$\alpha=M+Ne^{-\frac{B+E}{A}t},$$

where M and N are again the integration constants; to determine them we may again assume certain boundary conditions. For example, let the motion be considered from the moment when a certain disturbance, having deviated the ship over an angle $\alpha_0$, ceases to act. This gives for $t=0$

$$\alpha=\alpha_0;\ \frac{d\alpha}{dt}=\left(\frac{d\alpha}{dt}\right)_0$$

from which it follows that

$$\alpha_0=M+N;\ \alpha=\alpha_0+\left(\frac{d\alpha}{dt}\right)_0\frac{A}{B+E}\left(1-e^{-\frac{B+E}{A}t}\right) \quad (4)$$

$$\left(\frac{d\alpha}{dt}\right)_0=-N\frac{B+E}{A};\ \frac{d\alpha}{dt}=\left(\frac{d\alpha}{dt}\right)_0 e^{-\frac{B+E}{A}t} \quad (4')$$

* It is known that the intrinsic characteristics of an oscillating system are perfectly determined by the differential equation without the second member; the latter accounting merely for a particular solution depending upon the nature of an individual extraneous disturbance; for that reason the second member D may be disregarded in studying the oscillating features of the system. Some particular solutions corresponding to various second members are treated in the following.

Equations (4) (4′) show that both the deviation $\alpha$ and its angular velocity vary as exponential functions of time; the angle $\alpha$ gradually approaches a particular value

$$\alpha_t = \infty = \alpha_0 + \left(\frac{d\alpha}{dt}\right)_0 \frac{A}{B+E},$$

depending upon the initial angular velocity

$$\left(\frac{d\alpha}{dt}\right)_0$$

and A, B, E; the angular velocity thus gradually dies out. It is interesting to note that the action of angular velocity control denoted by the coefficient E, enters into the equations on equal terms with B.

The motion occurs in just the same way as if the water instead of opposing a resistance B to the angular motion of the ship, were a more resisting (viscous) fluid able to oppose a resistance B+E greater than B. Thus the angular velocity control of the character described accounts for an increased damping.

Steering by angular velocity alone is impossible because there is no directive force to stabilize the ship on a definite course; it has been shown that the ship deviates from her course without coming back to it; the apparent or effective resistance to turning is thus modified owing to the action of angular velocity control.

(*c*) *Angular acceleration control of angle of rudder.*

The third method is based upon the assumption that the angle of rudder is at all times proportional to the instantaneous value of angular acceleration, which can also be recorded by suitable means. Inserting the relation

$$\rho = p \frac{d^2\alpha}{dt^2}$$

into the general equation, we have

$$(A+F)\frac{d^2\alpha}{dt^2}+B\frac{d\alpha}{dt}=0,$$

where $F=pk$ is the term for intensity of the rudder action; the character of the motion is the same as in the second case, if we substitute $A+F$ for A and B for $B+E$.

The motion develops in just the same way as if, instead of the ship having moment of inertia A, its inertia were greater $(A+F)$, or smaller $(A-F)$, according to the way in which the accelerational control is acting on the rudder.

The remaining conclusions are the same as in the second case.

### 6. GENERAL CASE OF ANGLE OF RUDDER CONTROL.

Combining the three previous cases one has finally:

$$(A\pm F)\frac{d^2\alpha}{dt^2}+(B\pm E)\frac{d\alpha}{dt}\pm C\alpha=0 \qquad (6)$$

The double signs indicate the possibility of connecting the respective controlling elements in two different ways, of which only one is generally useful for steering. The general equation above contains all possible solutions for controlling the angle of rudder in proportion to the angle of yawing $\alpha$ and its time derivatives. The physical meaning of various controlling means may be classified, as follows:

(*a*) Compass Control C (otherwise: positional control of the angle of rudder) constitutes the directive couple for stabilizing the ship on her course. The timing is correct, if the directional angle of the rudder corresponds adequately to the deviation of the ship to check such deviation. The instability of control, or the wrong connection, occurs whenever the rudder, instead of checking the deviation, contributes to its further increase. Analytically

instability occurs whenever one of the roots of the equation is positive; in the case of imaginary roots when the real part of the imaginary quantity is positive.

(*b*) Angular velocity control (gyrometer control) accounts either for an increase (plus sign before E) or for a decrease (minus sign before E) of the natural damping due to the resistance of the water. For steering purposes it is, obviously, advisable to have a strongly increased damping, in order to approach the dead beat steering conditions; consequently the plus sign before E characterizing the intensity of the gyrometer action, which is of interest.

(*c*) Angular acceleration control F accounts, in similar way, for an increase (plus sign), or for a decrease (minus sign), in the apparent moment of inertia of the ship, characterized by the term $A \pm F$. A ship with a greater moment of inertia is less liable to be disturbed, but on the other hand, once disturbed the handling of the ship is more difficult. It appears, therefore, that in order to be able to use the accelerational control most effectively it should be introduced with the plus sign, when the disturbance first begins to act, and the action of the control should be reversed (minus sign) when the disturbance disappears. Consequently, the accelerational control is useful only when there is, so to speak, automatic selective discrimination between the beginning and ending of incoming disturbances.

In summing up, it may be stated that by proper application of the individual regulations denoted by C, E, F, we may obtain any steering characteristic which is desirable in an individual case.

Thus, for example, in the case where a very accurate dead beat course is to be steered, predominance should be given to the angular velocity control for increasing the damping effect in steering. In case of a rough sea, where a certain freedom of steerage direction is even advisable to avoid too violent action of the rudder, angular velocity

control may be decreased, while the compass control C may be given predominance and so on.

It is interesting to note that in the case where the angular velocity (or gyrometer) action is negative and equal to B, the total effective resistance (B−E) vanishes, and the angular motion of the ship becomes a sinusoidal undamped motion of a period depending upon the intensity of compass action C. Under such conditions the ship is able automatically to steer a sinusoidal, zig-zag course of any desired period.

The method above described for controlling the rudder is efficient from the standpoint of damping out the effect of a disappeared disturbance, but will not eliminate the effect of a steadily acting disturbing torque, such, for instance, as a steady wind. In order to steer the right course the ship must then have a certain amount of rudder consistent with the magnitude of the disturbing torque; on the other hand for no angular deviation, or no angular velocity, there is inevitably no angle of rudder, since the latter is supposed to be proportional to those quantities. It follows therefrom that the ship will be deviated from her course under the influence of such a steady disturbing torque D over an angle

$$\alpha_1 = \frac{D}{C},$$

representing analytically the particular solution of equation (3) corresponding to the constant second member D. Since the torque D may vary with the time, the angle $\alpha_1$ is also variable; the steered course becomes indefinite under those conditions.

This first class of steering devices, acting to regulate the angle of rudder, is impractical for this reason.

The following class of steering devices has no such disadvantage, while all the advantages inherent to the first class of devices are present in the second class now to be described.

## 7. SECOND CLASS OF AUTOMATIC STEERING DEVICES (RATE OF MOVEMENT CONTROL OF THE RUDDER).

In this second class of steering devices the angle of yawing $\alpha$ and its time derivatives do not control *the angle* of rudder, but *the rate at which this angle is varied.* Since for small angles, the restoring torque due to the rudder is proportional to the rudder angle, *we may also characterize this second class as devices in which the controlling elements*

$$\alpha, \frac{d\alpha}{dt}, \frac{d^2\alpha}{dt^2}$$

act *upon the variations of the restoring torque due to the rudder and not upon the final value of the torque, as in the first case.* The second case, consequently, is characterized by a relation of the form

$$\frac{d\rho}{dt} = m\alpha + n\alpha' + p\alpha''.^*$$

Substituting this expression in the fundamental equation of automatic steering, we have:

$$A\frac{d^2\alpha}{dt^2} + B\frac{d\alpha}{dt} + k\int (m\alpha + n\alpha' + p\alpha'')\,dt = D.$$

Differentiating and rearranging the terms, we have:

$$A\frac{d^3\alpha}{dt^3} + (B + kp)\frac{d^2\alpha}{dt^2} + kn\frac{d\alpha}{dt} + km\alpha = \frac{dD}{dt}.$$

In the case of a steadily acting disturbance

$$\frac{dD}{dt} = 0,$$

from which follows the remarkable result that such a disturbance has no influence upon the performance of the

* Where $m$, $n$, $p$ are coefficients for the degree of coupling between the speed of steering gear and angle, angular velocity, and angular acceleration of yawing, respectively.

device, depending solely upon the inertia A of the ship, the resistance B and constants, $m$, $n$, $p$, representing the intensities of the corresponding components of the control.

Such a device is responsive solely to the rate of increase or decrease in the action of the disturbance, and is not influenced by its actual magnitude; a device of this nature will steer the ship independently of any steady disturbances, and if the latter should vary, the steering will be adjusted automatically to the new value of the disturbance without affecting the correctness of the steered course.

The above result can also be understood from a more elementary discussion. Since in this case there is no rigid proportional relation between the elements of angular motion ($\alpha$, $\alpha'$, $\alpha''$) and the angle of rudder, the latter adjusts itself automatically until the motion is stopped; the ship is thus able to carry the required amount of rudder, being always on her right course.

The first member of the equation shows that the relations are totally different from those which were established in the case of the equation for devices belonging to the first class; namely,

1. The differential equation of motion is of the third order; in the first case it was of the second order.

2. None of the controlling means affect the angular inertia in this case; in the first case the accelerational control affects the inertia by the additional term F, whereas in this case it affects the resistance B by the term $kp$.

3. Velocity control by the gyrometer now appears as an independent term in the equation.

4. Only the positional control by the compass remains in the equation the same as in the first case.

A complete solution of the auxiliary equation of the third degree is not necessary for the purpose in view. The condition of stability of the ship on her course may be analytically expressed by the condition that none of the roots of the auxiliary equation may contain any positive

real part. If all three roots are real, the condition of stability requires that they must all be negative. If one root is real and there is a pair of conjugate imaginary roots, the real root must be negative and the real part of imaginary roots must also be negative.

M. A. Blondel has shown* that by applying the so-called Hurwitz Theorem of Analysis, the "stability of roots" of any algebraic equation of the $n$th degree can easily be established in the form of a series of simultaneous conditions, which he calls Hurwitz conditions of stability. Referring for more detail on the subject to the article by Blondel, we may simply say that in the case of an equation of the third degree of the form $b_0x^3+b_1x^2+b_2x+b_3=0$, the stability conditions are:

$$(1)\ b_1>0$$
$$(2)\ b_1b_2-b_0b_3>0$$
$$(3)\ b_3\,(b_1b_2-b_0b_3)>0$$

In view of the second condition the last condition is merely $b_3>0$; so that finally we have (1) $b_1>0$; (2) $b_1b_2-b_0b_3>0$; (3) $b_3>0$. In the case under consideration the auxiliary equation is: $Ax^3+(B+kp)\,x^2+kn\,x+km=0$.

The Hurwitz conditions for stability of an automatically steered ship on her course become then:

$$(1)\ (B+kp)>0$$
$$(2)\ (B+kp)\,kn-A\,km>0$$
$$(3)\ km>0$$

The last condition means simply that the compass (more generally the direction indicating apparatus) must be connected in "the right way" to the rudder, i.e., when the ship yaws to port the helm must be moved to starboard.

The first condition shows that stability may be upset if the accelerational control $kp$ is connected in the wrong direction so as to render $B+kp<0$.

* Journal de Physique, April and May, 1919.

*The second condition completely determines the stability of the ship on her course; it shows that for a sufficiently great value of n, representing the intensity of gyrometer control, the stable conditions can always be obtained, whatever the inertia of the vessel may be.*

It is interesting to note that the gyrometer action does not *add itself* to the resistance of the medium B, as in devices of the first class, but is *multiplied by it;* in other words, in the case of absence of velocity control ($n=0$), stability disappears because the second condition becomes $-A\,km$ which is always negative because both A and $km$ are essentially positive.

From the above relations we may conclude that accelerational control is of no special importance, and can be dispensed with; while velocity control (gyrometer) on the contrary constitutes the necessary and sufficient condition for stability of the steered course.

## 8. GENERALIZATION.

It has been shown that the two methods of rudder control are equally good, so long as there is no permanent disturbing torque acting on the ship. On the other hand, the conditions are totally different, if such a disturbance is acting; the "follow up" control of the rudder is then unable to maintain the ship on her course, while "the rate of angle control" is independent of any steady disturbances, but the ship is steered correctly provided the disturbance itself remains steady; if the disturbance is variable, it is only this variation that figures as the effective disturbance, and affects the instantaneous behavior of the device, but not its absolute value.

We may, therefore, proceed still further, and establish a third class of automatic steering devices in which the elements of angular motion of the ship control the acceleration of the rudder, and not its angle as in the first case,

nor its velocity as in the second case. Such a condition is expressed by a relation of the form

$$\frac{d^2\rho}{dt^2} = m_2\alpha + n_2\alpha' + p_2\alpha'',$$

which being substituted in the general equation for automatic steering, finally results in the differential equation

$$A\frac{d^4\alpha}{dt^4} + B\frac{d^3\alpha}{dt^3} + kp_2\frac{d^2\alpha}{dt^2} + kn_2\frac{d\alpha}{dt} + km_2\alpha = \frac{d^2D}{dt^2}.$$

Such a device will not be influenced either by permanent steady disturbances or by uniformly varying disturbances. It is doubtful, however, whether such a method is necessary or advisable for practical purposes in view of the character of the disturbances usually occurring. In general three different types of disturbing torques have to be taken into account: (1) steadily acting, (2) periodical, (3) accidental or irregular.

The first kind of disturbances is practically the only one that is of interest for big ships because they generally respond, not to individual disturbing impulse, but to the time integral of many impulses due to the waves whose action is thus cumulative with time; to the same category of disturbances belongs the wind, the strength of which does not, in general, vary abruptly; to the same kind of disturbing factors belong also the influence of trim, depending upon distribution of the cargo and similar permanent causes.

The second kind of disturbances is important only for smaller craft, responding instantaneously to each individual impulse or wave (which may become of importance for larger ships in a heavy sea); this disturbance constitutes what is generally called yawing among waves; in this case the periodicity of such a forced yawing is obviously the apparent periodicity of the waves, and the rudder can do

nothing to prevent it; in practice, in such cases intentionally loose steering is generally admissible in order not to over-regulate the rudder too much, which would only uselessly decrease the speed; this is also the only possible method in the case of automatic steering.

The third kind of disturbances is quite special and hardly likely to arise on seagoing ships, but occurs more frequently in aerial navigation (air pockets, bumps); the only way to overcome the effect of such disturbances rapidly is to diminish the time lag of the controlling transmissions to a minimum.

In view of the foregoing it is problematical whether the third class of rudder control can become of any practical interest. The first class of automatic steering devices (based on follow up steering gear) is decidedly objectionable in view of uncertainty of the steered course under the action of disturbances, although when no such disturbances are acting it yields quite a correct solution of the problem.

The devices of the second class seem to be the logical solution of the problem of automatic steering.

## 9. INFLUENCE OF TIME LAG OF TRANSMISSION UPON THE AUTOMATIC STEERING.

In the foregoing analysis it has been assumed that the restoring action of the rudder is instantaneously responsive to the elements of angular motion

$$\alpha, \frac{d\alpha}{dt}, \frac{d^2\alpha}{dt^2}$$

controlling the rudder. In reality it takes an appreciable time before such action begins; the delay is due to the transmission system possessing a certain amount of inertia, either mechanical or electromagnetic, lost motion, viscosity and similar causes. By perfecting the transmitting

means the lag may be reduced, but never totally eliminated. Any transmission system can thus be characterized in terms of its lag measured, for example, in seconds.

In other words, while the real motion at a certain moment $t$ is characterized by the angle $\alpha(t)$, angular vleocity $\alpha'(t)$, angular acceleration $\alpha''(t)$, the instantaneous rudder torque, existing at the same moment of time $t$, relates to the moments of past time, lagging by the intervals $T_1$, $T_2$, $T_3$ behind the compass, the gyrometer, and the acceleration-transmitting systems, respectively.

Considering only the second class of steering devices, the differential equation of angular motion becomes then:

$$A\,\alpha''(t)+B\,\alpha'(t)+k\int\Big[m\alpha(t-T_1)+n\alpha'(t-T_2)+p\alpha''(t-T_3)\Big]dt=D$$

Differentiating, we have:

$$A\,\alpha'''(t)+B\,\alpha''(t)+km\,\alpha(t-T_1)+kn\,\alpha'(t-T_2)+kp\,\alpha''(t-T_3)=\frac{dD}{dt}$$

The functional character of $\alpha$, $\alpha'$, $\alpha''$ etc. in respect to time is emphasized by the symbols $\alpha(t)$, $\alpha'(t)$, $\alpha''(t)$.

Since the lag in time may be made very small by suitable design and careful construction, the intervals of time $T_1$, $T_2$, $T_3$ are very small quantities in comparison with the period of yawing of the ship. This fact permits the use of well known approximations resulting from the expansion of Taylor's Theorem, and we thus have:

$$\alpha(t-T_1)=\alpha(t)-T_1\,\alpha'(t)+\frac{T_1^2}{2!}\alpha''(t)-\frac{T_1^3}{3!}\alpha'''(t)+\cdots$$

$$\alpha'(t-T_2)=\alpha'(t)-T_2\,\alpha''(t)+$$

$$\frac{T_2^2}{2'}\alpha'''(t)-\frac{T_2^3}{3'}\alpha''''(t)+\ldots$$

$$\alpha''(t-T_2)=\alpha''(t)-T_2\alpha'''(t)+$$

$$\frac{T_2^2}{2'}\alpha''''(t)-\frac{T_2^3}{3'}\alpha'''''(t)+\ldots$$

Neglecting the small quantities of the second and higher orders, we have:

$$\alpha(t-T_1)=\alpha(t)-T_1\alpha'(t)=\alpha-T_1\alpha'$$

$$\alpha'(t-T_2)=\alpha'(t)-T_2\alpha''(t)=\alpha'-T_2\alpha''$$

$$\alpha''(t-T_2)=\alpha''(t)-T_2\alpha'''(t)=\alpha''-T_2\alpha'''$$

substituting these values for $\alpha(t-T_1)$ etc. in the equation we have:

$$(A-kpT_2)\alpha'''+(B+kp-knT_2)\alpha''+$$

$$k(n-mT_1)\alpha'+km\alpha=\frac{dD}{dt}$$

and the Hurwitz-Blondel conditions of stability become:

(1) $B+kp-knT_2>0$

(2) $(B+kp-knT_2)k(n-mT_1)-(A-kpT_2)km>0$

(3) $km>0$

It will be seen, therefrom, that the conditions of stability are affected by delay in transmission, except in the case of the last condition, which remains unchanged.

The first condition shows that for a sufficiently great value of lag $T_2$ in gyrometer transmission the stability may be upset; for gradually increasing lag $T_2$ this will happen the sooner, the smaller the resistance of the medium to the angular motion of the ship.

The second condition shows that the time lag $T_1$ in compass transmission (more generally in transmission controlled by a direction indicating apparatus) diminishes the useful effect of the gyrometer control.

It is to be noted that the lag $T_1$ in accelerational control (if there be any) may even be beneficial, and contribute to stability of the steered course, by decreasing the effective angular inertia of the ship to a certain extent.

## CONCLUSION.

From the preceding study it is apparent that all possible methods of rudder control *do not actually anticipate* the disturbing angular motion, but merely utilize this motion at its beginning when its value is small for the purpose of impressing a properly timed reaction against it. It is therefore obvious that the disturbing angular motion must necessarily occur *before* any controlling means can be operative.

Starting from rest under the influence of an external disturbance, all elements of the angular motion (angle, angular velocity, angular acceleration and still higher time derivatives) appear simultaneously and at the limit $t=0$ it is impossible to answer the question "which is better for the control": angular acceleration, angular velocity or angle. Outside of this isolated point this indefiniteness disappears and the problem has a quite definite physical meaning insofar as each higher time-derivative passes through its characteristic stages (maxima, zero points, etc.) at an earlier epoch than a corresponding lower one.

Thus for example the angular velocity being analytically the time integral of angular acceleration it must necessarily depend upon the past values of acceleration which account for its present magnitude, in other words, in the physical succession of facts the angular acceleration always precedes the angular velocity and the latter precedes the angle of deviation. It is therefore of a decided advantage for the purpose of a sufficiently rapid control to utilize preferably higher time derivatives of angular motion.

But while the problem of anticipating the appearance of a disturbance theoretically is meaningless—it may have

a practical meaning if we agree to understand that by it considerable reduction of yawing in comparison with the manual control, now in use, can be effected.

The progress in this direction is clearly indicated and may be summarized as follows:

(*a*) Perfection of instruments detecting the angular motion ahead of the human eye.

(*b*) Perfection of methods utilizing these instruments for the purpose of a suitable timing consistent with the dynamical nature of angular motion.

The above paper must be considered as an attempt to solve the last mentioned problem.

# SHOP TOOLS FOR SPECIAL PURPOSES.

By Commander R. L. Irvine, U. S. N., Member.

The following figures and descriptions apply to shop tools or rigs designed and used by the Bethlehem Shipbuilding Corporation, Ltd. (Union Plant) in connection with Naval contracts executed by them.

Figure 1 shows an attachment to a twelve-foot Bement Boring Mill for machining propellers to a true screw uniform pitch. A heavy vertical spindle with a thread cut to the pitch of the propeller was rigged on the head of the boring machine and governed the vertical travel of the cutting tools. The cutting tools were carried in tool holders working on three horizontal arms under the boring head. These tools were fed by a screw thread in each arm, each operated by a ratchet lever at the end of the arm. The boring mill table, on which the propeller was secured, was rotated one-third of a turn (for three-bladed propeller), and then reversed by means of a chain of gears and belt-operated pulleys similar to the method used in driving the reciprocating bed of a planer. This same chain of gears served to drive through shafting, bevel gears, and worm and wheel, the nut operating on the threaded spindle, thereby giving the proper lift to the cutting tools as the propeller rotated. The spindle and tool head were counterbalanced overhead to provide smooth operation. As the spindle raised, the ratchets feeding the cutting tools were operated by means of small flexible wire cables attached to the spindle.

Figure 2 shows a Blade Setting Machine for blading Curtis turbine wheels. The turbine wheel disc was secured to the spindle of the machine, and as each blade was placed, the disc was rotated and an arm on the pedestal forced the root of

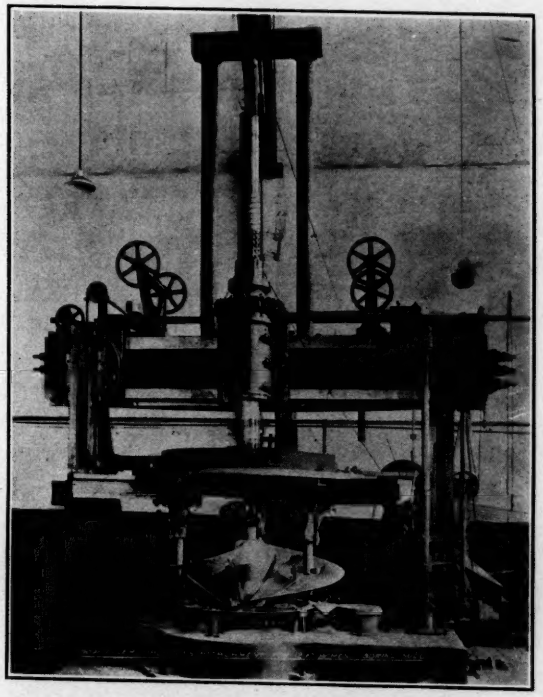


FIG. 1.—RIG FOR MACHINING PROPELLERS.

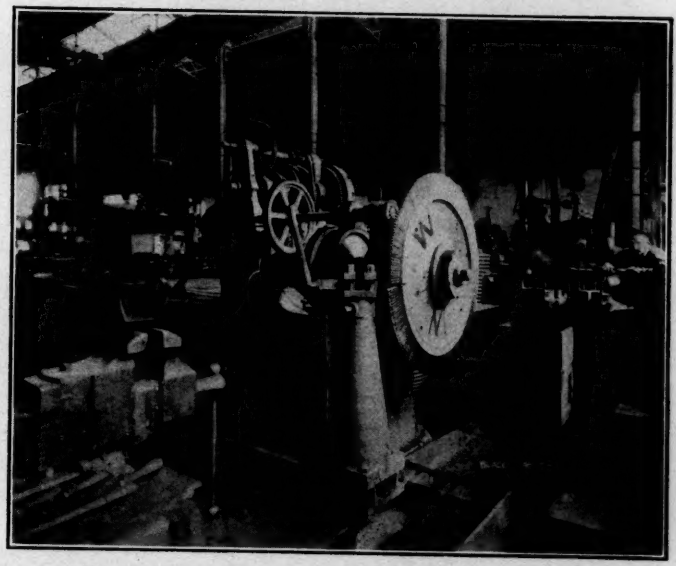


FIG. 2.—MACHINE FOR BLADING CURTIS TURBINE WHEELS.

FIG. 3.—TUBE MILLING MACHINE AND ANNEALING FURNACE.
YARROW BOILER SHOP.

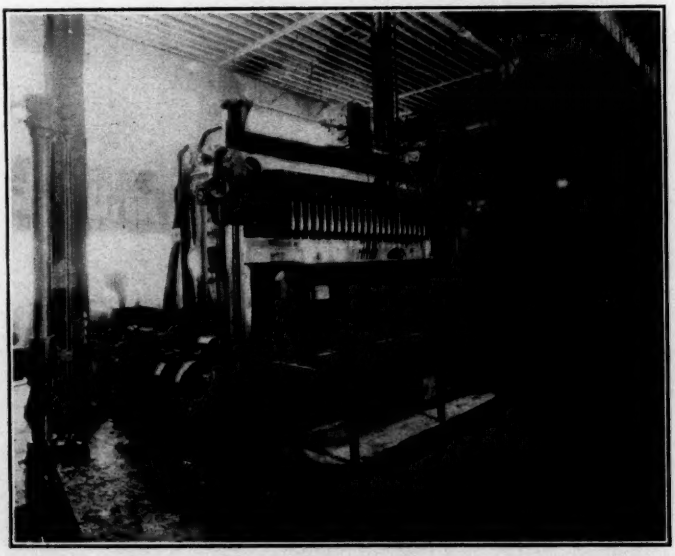


FIG. 4.—NEITZEL MULTIPLE DRILL.

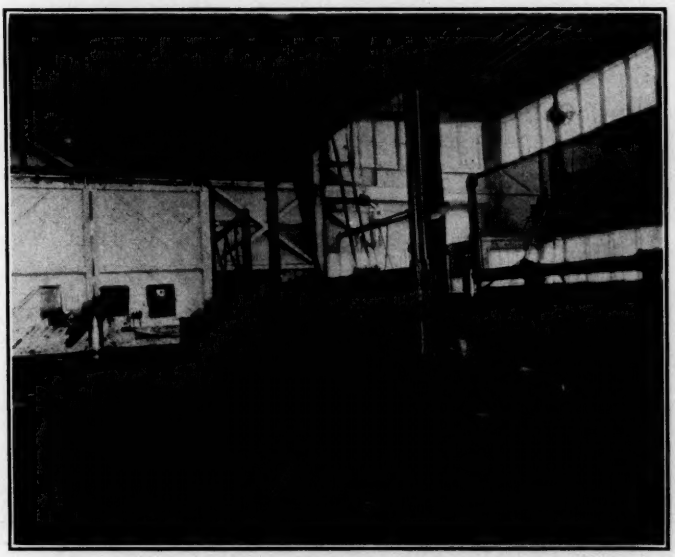


FIG. 5.—CHAPMAN AIR DRILL.
PNEUMATIC DRILLS RIGGED FOR DRILLING AND REAMING BOILER TUBE SHEETS.

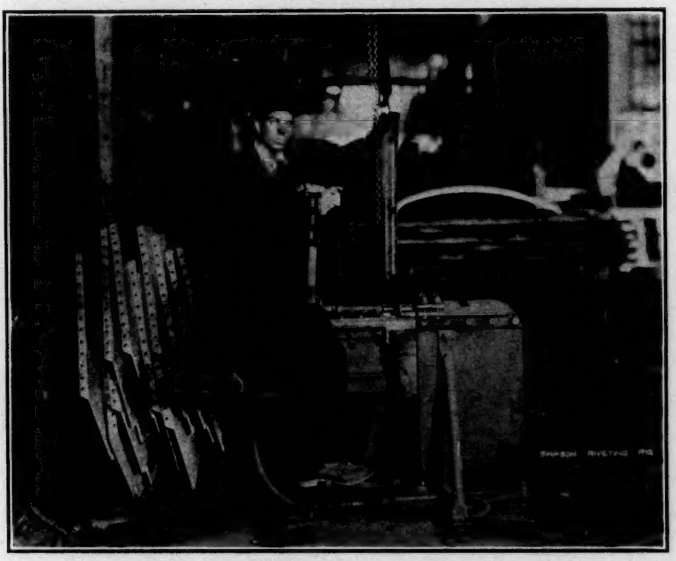


FIG. 6.—RIVETING RIG FOR BOILER CASINGS.

the blade firmly against the preceding blade. The gears of the tool were driven continuously by individual direct motor drive. The connection between the driving gears and the spindle carrying the turbine wheel was by means of a double-cone, wood-faced friction clutch operated by the hand lever shown in the figure on the front of the head stock. Forward and reverse motions of the spindle were secured through the clutch by the alternate positions of the operating lever, thereby forcing the blading on the turbine wheel in both directions from the entering slot, and effecting double speed in the blading operation.

Figure 3 shows a Tube Milling Machine and Annealing Furnace for rounding the walls at the ends of boiler tubes and for annealing the ends of these tubes. This equipment was found necessary in connection with the manufacture of Yarrow boilers for destroyers and scout cruisers. In the early stages of the work, the tubes that were furnished were found to be too brittle to stand expanding and belling, and it was necessary to anneal the ends. For this purpose, the tube milling machine and annealing furnace illustrated were designed and installed in the corner of the boiler shop, near outside tube racks where the tubes were unloaded from cars on a spur track. This tube milling machine and annealing furnace were supplemented with a polishing machine for polishing the tube ends. This polishing machine consisted of a table with a motor-driven grinding wheel at each end, against which the tubes were pushed by hand. The arrangement shown in the figure was later improved upon. The tubes were passed in from the tube racks by hand, to a traveling, endless chain which carried the ends through the two sections of the annealing furnace, which was gas-fired. The speed of this chain could be so regulated as to allow the ends of the tubes to heat the required amount while passing through to the lime pit at the end of the machine, where they were covered and allowed to cool slowly. From this lime pit, the tubes were passed to another endless chain which fed

them to the tube end miller. This tool had end millers adjustable to desired length of tube. These were pneumatically operated and milled the tube to length, and rounded the walls at the tube ends, the tube being clamped during the operation by pneumatic vises. When the millers were released, the tubes were ejected automatically by pneumatic power, and passed to chain belts which carried them to the polishing wheels where both ends were polished, and the tubes were ready for shaping and installing.

The tubes entered the drums of the boiler normally and it was, therefore, necessary to bend the ends to suit. This was done by hydraulic pressure on a specially designed table, by means of a ram and dies. The tubes were then placed in racks in the shop, ready for use. The tube sheets were drilled by the Neitzel multiple drills, Figure 4. These drills had twenty-two spindles each, which could be raised and lowered as a set, by the operator. The tube sheet to be drilled was assembled with its wrapper sheet and heads with temporary bolts, and the drum placed in position, in saddles on the table of the drill. A drum arrangement with adjustable stops, shown on the extreme left corner of the drill, automatically rotated the boiler drum the required spacing as the spindles were lifted after drilling a row of holes. High-speed drills of uniform length were used, and cutting compound was fed to each drill by individual pipe lines, by means of a small centrifugal pump taking suction from a sump tank under the base of the drill.

Figure 5 shows a Chapman Air Drill rig set up for reaming and counter-boring boiler tube sheets. An "I" beam carried on two upright columns and adjustable vertically by means of threaded spindles operated through bevel gears, by a lever on one pedestal, carried enough pneumatic drills on rollers to suit the work. The drills were moved independently by the operators. The tube sheets rested in saddles which could be rotated to present the rows of tube holes to the drills as required. The saddles rested in a framework on wheels riding

on tracks so that the carriage could be run out from the tools. Cutting compound was fed by flexible tubing to each drill. The compound drained to a sump tank under the floor, from whence it was returned by a small centrifugal pump.

Figure 6 shows a Simpson Riveting Rig designed for light riveting. It shows an operator and helper riveting a section of boiler casing. This rig consisted of a frame of boiler plate with a slot for the work to pass through. The top of this frame carried the riveting tool, which was an ordinary air hammer, with a lever for forcing it up to the work. The air valve was operated by a foot pedal as shown.

# NOTES.

## ROLLED PRODUCTS OF 99 PER CENT NICKEL.

### An Open-Hearth Treatment Which Makes Possible the Rolling of the Same Shapes Now Produced in Mild Steel.

By Edwin F. Cone.

The rolling of nickel, 99 per cent pure, into the various shapes into which mild steel is rolled is an achievement of no small interest to the metal-working trade. Not only are rolled nickel products possible, but forgings have been attempted with success. In addition, fabrication has been accomplished with no more difficulty than in the case of mild steel. An interesting feature of this development is that the same apparatus is used for working the nickel as is used for the steel. It is possible to roll steel bars and then immediately to introduce pure nickel billets or ingots into the rolls.

The Nickel Alloys Co. is now rolling steel into bars and shapes, and is converting nickel, 99 per cent pure, into bars, plates, sheets and other forms on an extensive scale. Dr. Charles T. Hennig, who is responsible for this development in nickel products, has experimented for many years in making nickel malleable enough to be rolled or forged into various shapes. He considers that the objects he has sought are now fully attained, though the rolling of pure nickel had long been considered impossible. The company's plant is located at Hyde, a small town in Clearfield County, Pa. It had previously been operated as a rolling mill. In 1916 Dr. Hennig obtained possession of it. After completely rehabilitating it and installing new equipment, he continued the rolling of steel while the development of the commercial production of pure nickel was under way.

Because of the non-corrodibility of pure nickel and its antiseptic properties, those interested predict its extensive use in many industries. It is especially suitable, as insuring easy sterilizing, in dairy machinery of all kinds; in dye house equipment, where acid and alkaline solutions are used; in gas and oil engines, where extremely high temperatures prevail; in marine installations where parts come in contact with salt water; in pickling and chemical works; in power plants and mining equipment. A large use for it as milk cans is expected. The high scrap value of the metal is an important commercial factor. Unusual strength and durability, affording lighter weights for specified purposes, and the fact that non-corrodibility insures longer usefulness, are cited as offsetting the higher cost.

The new product has been given the trade name Niaco nickel, and the following properties are claimed:

It is 99 per cent pure, is malleable and is the purest nickel commercially produced.

It has great resistance to corrosion caused by acid fumes and acids, by alkalies, superheated steam, etc.

It oxidizes little at high temperatures.

It can be welded to iron, steel or to itself.

It is white in color, resembling silver.

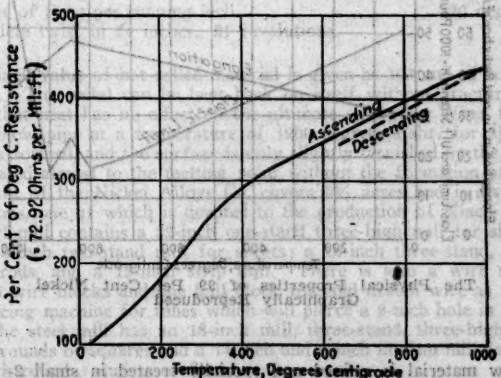
It has a specific gravity of 8.871 at 0 degrees C.

The average chemical composition is as follows:

| | Per Cent. | | Per Cent. |
|---|---|---|---|
| Carbon | 0.025 | Silicon | 0.155 |
| Manganese | Trace | Iron | 0.60 |
| Phosphorus | 0.015 | Copper | 0.12 |
| Sulphur | 0.025 | Nickel (plus cobalt) | 99.06 |

The melting point is 1485 degrees C. The coefficient of expansion from 25 degrees C. to 200 degrees C. is 0.0000125 and from 25 degrees C. to 300 degrees C. it is 0.0000135. Tests on the resistance of this metal to corrosion have shown the following results:

| Reagent | Hours. | Loss Per Cent. | Per Cent Loss Per Hour. |
|---|---|---|---|
| Nitric acid, 1 per cent | 462 | 1.227 | 0.002655 |
| Sulphuric acid, 1 per cent | 462 | 0.526 | 0.001128 |
| Hydrochloric acid, 1 per cent | 462 | 0.4282 | 0.000849 |
| Salt solution, 1 per cent | 462 | 0.0508 | 0.000110 |
| Potassium hydrate, 1 per cent | 462 | 0.05123 | 0.000111 |
| Water | 462 | 0.00000 | 0.000000 |
| Potassium cyanide | 473 | 0.423 | 0.000913 |
| Concentrated salt | 462 | 0.1198 | 0.000259 |



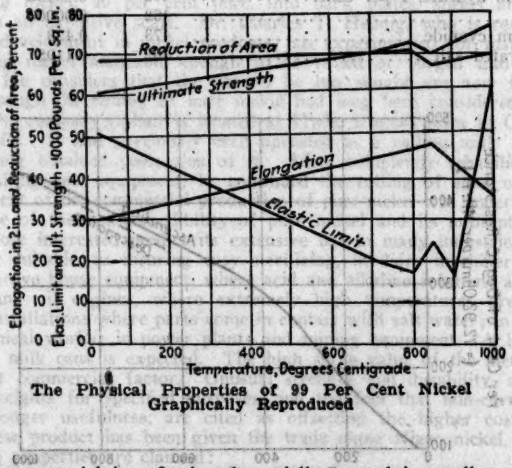

The Specific Resistance of Niaco Nickel Shown Graphically

The specific resistance of this pure nickel has also been determined as follows:

| Ascending Temperature. | Resistance in Ohms. | Specific Resistance. | Per Cent. |
|---|---|---|---|
| 25 | 0.065 | 72.92 | 100.00 |
| 200 | 0.123 | 138.00 | 190.21 |
| 400 | 0.191 | 214.40 | 294.00 |
| 700 | 0.239 | 269.00 | 367.00 |
| 1,000 | 0.284 | 319.00 | 437.00 |

A diagram showing these values graphically is reproduced.

For the crude nickel obtained in the open market as raw material, Dr. Hennig has developed a special treatment preliminary to rolling it into the many shapes produced at the Hyde plant. Striking malleability, under all conditions of heat and cold, has been secured. The writer has seen a 4-inch section of a 1-inch round rolled bar of this metal flattened cold by upsetting under a 2500-pound hammer until it was about 3½ inches in diameter and ⅜ inch thick, with no cracks or seams apparent. Also a 1-inch bar has been forged down hot to about ½ inch and then flattened under the hammer until cold. This was then reheated and folded over on itself and again flattened under the hammer, until there were 128 folds in the resultant piece which showed only a few evidences of cracks or brittleness. It is Dr. Hennig's claim that this is not possible with ordinary commercial pure nickel.




The Physical Properties of 99 Per Cent Nickel Graphically Reproduced

The raw material is refined and specially treated in small 2- to 3-ton open-hearth furnaces, specially designed by Dr. Hennig. The hot metal is poured into ingot molds such as are used in making steel, at a temperature of approximately 3200 degrees Fahr. Various sizes of ingots are cast, the largest at present being one ton. These ingots are later broken down under hammers or in rolls after the usual preheating. To insure a perfect surface on the product, the ingots and sheet bars are always carefully machined.

Sheets are rolled down in packs of 8 to 32 sheets to a thinness of 0.001 inch. A large powder company is using this very thin metal in a special cartridge for smokeless powder.

Besides the properties given in the earlier part of this article, the following static properties are furnished as characteristic of Niaco nickel:

| | Elastic Limit Lb. per Sq. In. | Yield Point Lb. per Sq. In | Ultimate Tensile Strength, Lb. per Sq. In. | Elongation in 2 in., Per Cent | Reduction of Area Per Cent |
|---|---|---|---|---|---|
| Heated to 700 degrees and hot rolled to 1-inch rods..... | 41,000 | 56,000 | 84,000 | 31.0 | 70.2 |
| Cold drawn 1-inch rods..... | 54,000 | 100,000 | 101,000 | 10.9 | 65.4 |
| Wire 0.500 in. dia..... | 62,300 | ..... | 80,900 | 34.0 | 71.0 |
| Wire 0.186 in. dia..... | 89,900 | ..... | 105,200 | 11.5 | 59.2 |
| Wire 0.062 in. dia..... | 114,500 | ..... | 127,500 | 8.0 | 60.5 |
| Wire 0.031 in. dia..... | 128,000 | ..... | 140,000 | 6.0 | 42.4 |

A graphic representation of these values is reproduced. The high elastic ratio in some of these results is noteworthy. Other physical tests are furnished as follows:

*Transverse test on a 1-inch diameter hot rolled bar:*

| | |
|---|---|
| Elastic limit | 27,500 lb. fiber stress |
| Utimate strength | 84,300 lb. fiber stress |
| Modulus of elasticity | 16,500,000 lb. fiber stress |

*Torsion test of a 1-inch rod, hot rolled:*

| | |
|---|---|
| Elastic limit | 12,250 lb. fiber stress |
| Ultimate strength | 80,800 lb. fiber stress |
| Degree of twist per running inch | 320 degrees |
| Complete twist in 24 inches, 21 revolutions. | |

The Brinell value of hot rolled material is given as 103 and of cold rolled up to 195. The metal can be bent back on itself without fracture, and it is stated that heat has no effect on the ultimate strength. When heated in air by a blow pipe at a temperature of 1900 degrees Fahr. for 3½ hours, no scale is formed, and the surface is only slightly tarnished by the heat. It can be heated almost to the melting point without the formation of scale.

The plant of the Nickel Alloys Co. covers 5½ acres and is divided into two portions, one of which is devoted to the production of Niaco products. The nickel mill contains a 22-inch one-stand three-high mill for sheets and rods; a 20-inch two-stand mill for sheets; a 12-inch three-stand, two-high mill for rods, and a 12-inch strip mill. There is also a wire mill containing 15 wire blocks and one bull block for cold drawn wire as well as a large piercing machine for tubes which will pierce a 2-inch hole in a 26-inch billet. The steel mill has an 18-inch mill, three-stand, three-high for 1½ to 4-inch rounds or squares and a 14-inch and 9-inch tandem mill for ¼-inch to 1½-inch rounds, squares or flats.

Besides Niaco nickel, the company produces nickel coated steel sheets or other products by rolling nickel sheets in conjunction with steel billets. By placing the nickel on one or both sides of the steel billet or slab the desired product is obtained by welding. It is also possible to produce a highly polished nickel coated steel sheet.

Niaco tubes have been produced for service in locomotive boilers, and one railroad has obtained interesting results in such service.

It has been found practically impossible to break sheets of moderate thickness by bending. A sheet 15 feet by 52 inches by 3-16 inch thick is exhibited as the largest that has been made from pure nickel. In the passage through the rolls no scale is given off, and the working of the metal hot is a pleasant sight. Niaco nickel can be quickly annealed by heating to a yellow heat and plunging into cold water. It will then become as soft as copper. If plunged into liquid air it still retains its malleability, while copper and some other metals become so brittle that they will disintegrate or become granular.

George P. Bassett is president of the Nickel Alloys Co., Dr. C. T. Hennig is vice president, and T. J. Patton, secretary and treasurer. The directors are George P. Bassett, Dr. C. T. Hennig, W. J. Webster, president of the Atlas Powder Co.; Leland Lyon, treasurer of the same company, and Leonard Richards, president of Richards & Co., New York. The general offices are at 740-746 Henry W. Oliver Building, Pittsburgh.—"Iron Age," June, 1920.

## "VIBRAC" STEEL.

The production of alloy steels having various desired properties offers plenty of scope for the ingenuity of the research departments of our great iron and steel works. The different rare metals such as chromium, nickel, tungsten, and vanadium have a marked effect upon the properties of iron even when they are alloyed with it in comparatively small quantities, and sometimes even a slight deviation in composition from a previous form of "alloy-steel" marks a very great change in its mechanical properties. It is of interest to note that a new form of steel, to be known as "Vibrac," has been produced by Armstrong, Whitworth & Co., as a result of research work which has been carried out at their Openshaw works at Manchester. It is stated that this steel has the remarkable property of never tempering brittle, and of being absolutely consistent in its behavior under any normal treatment. One of the most serious disadvantages of many forms of high tensile steel is the brittleness which develops as a result of tempering or other heat treatment; moreover, this brittleness is not uniform, but may vary within wide limits for the same steel. If, as is claimed, the new material not only yields a high impact value after tempering, but gives consistent values throughout its mass even when large forgings are being dealt with, it should prove of extreme value in shipbuilding and engineering construction—"Shipbuilding and Shipping Record," February, 1922.

## A NEW MAGNESIUM METAL.

At a meeting of the Lancashire Branch of the Institution of British Foundrymen, Mr. J. D. Paton described the characteristics and properties of a white metal made from magnesium, which, he said, would assume great importance in the future in the production of light castings. It was much lighter than aluminum—the weight ratios being approximately: Magnesium 1, aluminum 2, iron 6—and could be substituted for aluminum in almost all its uses. Being compelled to find such a substitute during the war, the Germans resorted to magnesium because experience had already

shown that that metal would partly meet the requirements. The developments which took place in Germany resulted in an output of the new metal which was equivalent to the pre-war world consumption of aluminum—between 20,000 and 30,000 tons. The casting of magnesium required care, highly dried moulds and cores being essential; but the moulder could readily acquire some knowledge of the physical and mechanical characteristics of the metal, and with careful handling, correct heating, and dry floors and moulds, castings could be produced without trouble or danger of explosion. About 500 pounds of coke would smelt 1,000 pounds of metal, the heat consumption being only two-thirds of that in the case of aluminum. For thick-walled castings a pouring temperature of 650 degrees to 680 degrees C. was required, and for thin-walled castings 700 degrees to 750 degrees C. When cast at the correct temperature in a perfectly dry mould, the metal was of a silvery color. It was very ductile, and would be used in innumerable ways, as, for instance, in aircraft; in the electrical industry, in all cases where aluminium was now used; and in the textile industry, where parts of machinery now made of cast iron and weighing several hundredweights could be made of magnesium, the consequent reduction in weight leading to economy in power. A reduction might also be effected in transport costs by the use of the metal. Magnesium was impervious to the action of alkalies, but was attacked by acids, and might therefore be applied in the soda industry.—"Mechanical World," December, 1921.

## LIGHT METAL PISTONS.*

By Prof. Becker, Dr.-Eng.

In the recent competition for aluminum pistons held by the German Government Department of Aircrafts and Automobiles, very valuable information regarding the utilization and fashioning of light metal pistons was obtained, all of which should prove of considerable value to the trade.

The investigations and tests were made in the Testing Laboratory for Automobiles and Trucks, at the Technical University, Charlottenburg, on altogether thirty-two sets of light metal pistons composed of sixteen different aluminum and magnesium alloys. For the purpose of comparison with these, two sets of cast iron pistons and one set of copper pistons were also tested. To carry on these tests there was utilized one 45-horsepower truck motor and one 10-30-horsepower automobile motor, and also a number of specially erected other types of equipment; first the mechanical and operating value of the different light metal pistons, as well as the changes which appeared in the operating efficiency of the motors, were systematically sought, and there was also systematically sought material suitable for the manufacture of pistons having characteristics decisive for piston operation of the highest efficiency.

The particular thermic advantages of the light metals, aluminum and magnesium almost entirely remove an important disadvantage in the design of truck motors and, in fact, all light motors. This disadvantage consists of the fact that mainly only the cylinders of the motors can be cooled directly and immediately by means of water or air, while the uncooled walls of the combustion chambers are exposed to great over heating by the gases, when in operation. To these uncooled areas belongs, in

* Reprint from the "Zeitschrift für Metallkunde," October issue, 1921. (Edition of the German Engineers' Club, Berlin.)

first importance, the considerable area of the piston surface. It is desirable to cool these uncooled areas in such a way as to enable the heat taken on by them to be conducted on through the material and delivered to the adjacent approximately cooler walls. Therefore this makes it necessary to consider high heat conductivity of piston material as of prime importance, but at the same time it is also necessary to have piston material of low specific weight in order to permit sufficiently thick walls in the body of the piston to enable a flow of heat through as big a cross-section as possible.

These light metals compared to iron have a very good heat conductivity and there is, of course, a considerable difference in the weights in comparison with iron. In truck motors, however, by making use of light metal pistons, it is possible to attain an even heat in those walls bordering the combustion chamber which are both quickly and slowly cooled. This enables the compression to be considerably increased, thus increasing the efficiency and decreasing the gas consumption. The best motor performance with the smallest consumption of gas was attained in the two test motors with 5.7 times compression. In these two test motors, the best two light metal pistons, as compared with cast iron pistons, showed 21 per cent greater performance with an accompanying reduction of 20 per cent in the consumption of gas. Aside from this very important dynamic and economic gain, light metal pistons enable a reduction in the loss of heat through cooling water of 35 per cent and a reduction of 20 per cent in exhaust gas temperature, *i. e.*, far above 100 degrees C. Therefore, by using light metal pistons, the cooling space of the motor can be made smaller and the danger of over heating in the exhaust valve is considerably lessened. The light metal pistons compared with cast iron pistons showed a saving of approximately 50 per cent in the consumption of lubricants.

The leaping progress attained in motor construction through the application of these highly efficient light metal pistons is based on the technical exploitation of those advantages of the material used which were never considered before. In cast iron pistons the temperature rises very strongly from the middle of the piston shaft to the center of the piston surface. Against this, the copper pistons have, on account of their very high heat conductivity, an even and lower temperature in all parts of the piston surface, which only has an unimportant falling off even down to the lower end of the piston shaft. The temperature curve of the light metal rests between that of the cast iron pistons and the copper pistons and approaches very close, in the higher grade alloys, to the ideal copper curve. The differences in the temperature declines of the sixteen tested alloys were quite considerable and will be further characterized and shown in the general and more extensive summary to be published.

In investigations which were made, the influence on the temperature of the carbon which is formed in operation on the piston surface, was also firmly determined. With cast iron pistons and also with all the light metal pistons the carbon results in a considerable raising of the temperature in the body of the pistons. The raising in temperature between clean pistons and carbon-covered pistons is approximately the same for all the piston materials tested. Although the temperatures of all types of carbon-covered pistons increase, the advantages of light metal pistons, in comparison with cast iron pistons, are fully maintained in their cooling.

Cast iron pistons absorb very much heat from the gases touching the piston surface, but on the other hand the heat absorption of the same size copper pistons, at the same temperature declines in the gas and the cylinder cooling water, is hardly half so large as the heat absorption of the cast

iron pistons. The heat absorption of light metal pistons lies about midway between the heat absorption of cast iron and copper and is 30 per cent lower than the heat absorption of cast iron.

The marked valuable characteristics of light metals are, however, only exploitable in motors when these materials are given, through suitable alloying and special constructing processes, good operating characteristics and resistance against heavy wear. The pistons with the highest "ball-impact-hardness" had the best qualities for operation. Hereby the light metals are divided into two groups. In one group lie all alloys with aluminum content; in the second group lie the magnesium alloys without aluminum content. As soon as magnesium is alloyed with amounts of aluminum worth mentioning, the alloy moves into the first, or aluminum group. The minimum hardness for a good piston operation for the aluminum group was 100—and for the aluminum-free-magnesium group 60. The aluminum alloys which took part in the competition had hardness values ranging between 70 and 120; aluminum-free-magnesium alloys had hardness values of 48 to 67. The value of the various pistons in operation varied according to the alloys used. The best values, with regard to hardness and also piston operating efficiency, were shown by the aluminum alloys with 12 to 15 per cent copper content which were cast in iron forms and with an iron core, and by the Elektron alloys with 13 to 15 per cent copper content stamped in dies. These alloys indicated the best thermic qualities as well as, correspondingly, the best operating values, in the motor trials.

The stroke surface of the 12 per cent aluminum without copper being large, therefore the wear is considerably greater than as compared with the piston made of Elektron with the copper content. These pistons showed, as compared with the hardest aluminum alloys, the same value in operating efficiency.

In order to keep the ring groove sides as small as possible to prevent the breaking through of the grooves, it is necessary to use the lowest possible compression rings. The upper ring should not slide over. The boxes for the piston bolt eyes must be amply designed as otherwise an expansion of the bolts will set in caused by the rapid succession of alternating stresses, and loosen the box position. The piston surface and the upper part of the piston shaft must be amply designed for a sufficient heat-flow-cross-section. The shape of the piston surface can be either vaulted, straight or conical, but it must be so designed that the charge remains pent as long as possible, and that the combustion, introduced at the ignition point, can progress through the middle of the charge in the shortest way. With the present methods of designing motors, these requirements cannot always be fulfilled, because the greater stroke in the combustion chamber necessary for higher compression can, in many cases, be reached, in light metal pistons, only through elevating the piston surface. Otherwise, the cylinder must be accordingly changed in design.

## INTERNAL-COMBUSTION ENGINEERING.

### An American Solid-Injection Diesel Engine.

Worthington Airless-Injection Oil Engine. The writer uses the expression "airless injection" to indicate its difference from the air-blast injection of the Diesel engine, although the expressions "solid injection" and "mechanical injection" have hitherto been more common.

In the new Worthington engine fuel oil is delivered to the engine by a service pump operated by an eccentric on the crankshaft.

The engine is of the valveless two-cycle crosshead type with a separate compartment between the cylinder and the crankcase for scavenging air.

One of the distinguishing characteristics of this engine is the use of a divided, or two-part, combustion chamber. The function of this divided combustion chamber is to reduce explosive pressures and to create a condition of air turbulence in the main combustion chamber during the combustion period. Shortly before top dead center, fuel is injected through the spray valve in an atomized condition directly into the smaller of the two compartments, known as the "injection chamber," this being located above and directly in communication with the main combustion chamber or cylinder clearance space. Ignition of the fuel is from the heat of compression and the period of injection is so timed that the partial burning of the fuel charge in the injection chamber produces sufficient pressure to start the flow of the main part of the fuel charge down into the cylinder until the jet of gaseous oil and air from the injection chamber attains considerable velocity, producing a turbulent condition in the cylinder just as the piston starts on its downward stroke. This is accelerated by the downward motion of the piston.

Combustion then takes place in the lower chamber or cylinder, under conditions closely approximating the air-injection Diesel engine, the resultant expansion of gases driving the piston down on its power stroke, after which the cycle described above is again repeated. The pressure from the fuel pump is high, but lasts only during injection about 15 degrees of crank angle. It will be noted that the time and rate of combustion are independent of time and rate of pump injection.

One of the refinements of the Worthington engine consists in providing a scavenging-air connection on the base of the engine which may be piped to the outside of the engine room. By this means a supply of pure, fresh air may be had for scavenging purposes in places where the air of the engine room is charged with dust or explosive vapors.

The fuel-injection pump is of the unpacked type, and as will be seen from Fig. 11, is driven by an eccentric (17) mounted on the crankshaft. Each cylinder has a separate and independent pump, complete in all its details. By avoiding the use of packing from the pump plunger, danger of the plunger sticking is eliminated, and at the same time leakage past the plunger is considerably less than in the usual type of packed pump. The pump body (24) is made from a solid block of steel, designed and constructed with great care, and the plungers and valves are assembled in this body of a unit.

Actuation of the pump plunger is by the eccentric (17), the crosshead (19) engaging the tappet (21), which in turn pushes the plunger upward on its injection stroke, the plunger being returned to its at rest position by the spring as the eccentric passes its top position. The actual instant at which the pump plunger begins to move upward determines the instant of injection of oil into the injection chamber.

For control, a centrifugal governor is fitted, acting through a series of rods and levers to rock the control shaft (10). This shaft is turned eccentric at a point in line with each fuel-pump plunger, and a so-called "by-pass lever" (9), actuated at one end by the pump plunger, rests at the other end on the eccentric part of the control shaft through the medium of an adjustable set screw. As the end of the by-pass lever moves upward with the pump plunger the by-pass tappet (9) and the plunger (7), resting on the lever, will move upward also, closing the gap between its

upper end and the suction or by-pass valve (9). When contact is established at this point the continued upward movement of the plunger will open the valve, permitting the oil to by-pass back to the fuel reservoir instead of being forced through the spray valve, and thus ending the injection or effective pump stroke. It will be seen that by rocking the control

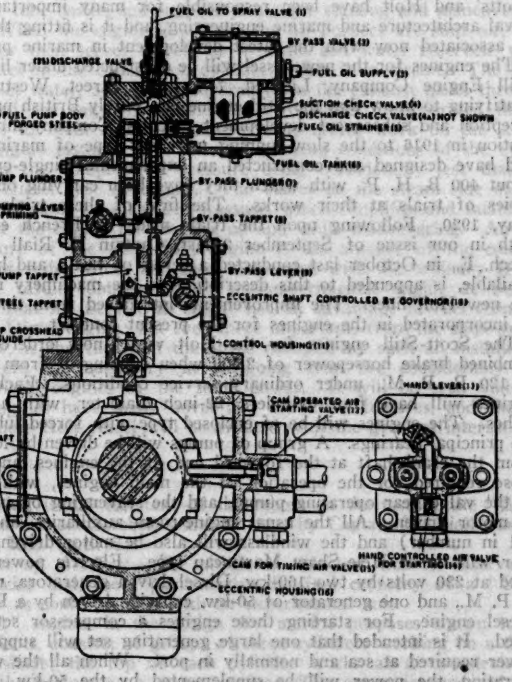

FUEL OIL TO SPRAY VALVE (1)
BY PASS VALVE (2)
(3) DISCHARGE VALVE
FUEL OIL SUPPLY (5)
SUCTION CHECK VALVE (6)
(8) FUEL PUMP BODY FORGED STEEL
DISCHARGE CHECK VALVE (6A) NOT SHOWN
FUEL OIL STRAINER (7)
FUEL OIL TANK (16)
(9) FUEL PUMP PLUNGER
BY-PASS PLUNGER (9)
REMAND PUMPING LEVER FOR PRIMING
BY-PASS TAPPET (9)
(11) FUEL PUMP TAPPET
BY-PASS LEVER (9)
ECCENTRIC SHAFT CONTROLLED BY GOVERNOR (15)
(12) HARDENED STEEL TAPPET
(13) FUEL PUMP CROSSHEAD & GUIDE
CONTROL HOUSING (11)
HAND LEVER (17)
CAM OPERATED AIR STARTING VALVE (17)
(14) CRANK-SHAFT
(15) FUEL PUMP ECCENTRIC
HAND CONTROLLED AIR VALVE FOR STARTING (18)
CAM FOR TIMING AIR VALVE (17)
ECCENTRIC HOUSING (15)

FIG. 11.—SECTION OF WORTHINGTON ENGINE FUEL PUMP.

shaft (10) one way or the other from its normal running position, the instant of opening of the suction valve by the by-pass plunger may be retarded or advanced, thus increasing or decreasing the effective pump stroke and feeding more or less oil into the cylinder, according to the load requirements. Also, by regulation of the set screw in the end of the lever (9), the instant of by-pass opening for each cylinder may be individually adjusted for purposes of initial setting and load balancing (Motorship, vol. 6, no. 11, pp. 872-875, 10 figs., d).—"Mechanical Engineering."

## A NEW DEVELOPMENT IN MARINE PROPULSION.

The contract which was placed recently with Messrs. Scotts' Shipbuilding and Engineering Company, Limited, Greenock, for the construction of a 400-foot twin-screw vessel for Messrs. Alfred Holt and Co., of Liverpool, has attracted a considerable amount of attention in engineering and shipowning circles throughout the country on account of the departure made in connection with the propelling machinery for which Scott-Still engines have been adopted. In an association extending back to 1865, the firms of Scotts' and Holt have been responsible for many important advances in naval architecture and marine engineering, and it is fitting that they should be associated now with the latest development in marine propulsion.

The engines for the new vessel will be constructed under license from the Still Engine Company, Limited, 7 Prince's Street, Westminster. It is gratifying to know that the Still engine is a purely British production in its inception and subsequent development. Scotts' Company directed their attention in 1916 to the slow-running merchant type of marine Still engine, and have designed and constructed an experimental single-cylinder unit of about 400 B. H. P., with which they have been carrying out an extensive series of trials at their works. The first of these trials was made in May, 1920. Following upon the trials made by French engineers, dealt with in our issue of September 2 last, Captain H. Riall Sankey, M. I. Mech. E., in October last conducted a series of trials, and his report, now available, is appended to this description of the machinery installation for the new Holt liner. The improvements introduced from time to time will be incorporated in the engines for the present contract.

The Scott-Still engines for the Holt vessel now ordered will have a combined brake horsepower of 2,500 when running at from 115 R. P. M. to 120 R. P. M., under ordinary service conditions. Each set of main engines will have four cylinders, 22-inch diameter, with a stroke of 36 inches. The engines will be of enclosed type with forced lubrication to all the principal bearings. A group of pumps will be driven by levers and links from the crankshaft at the forward end of the engines, and will embody those required for the operation of the main engines, with the exception of the valve gear operating pumps, and the scavenging blowers which will be motor driven. All the usual engine-room auxiliaries, and the winches (11 in number) and the windlass will also be motor driven. The steering gear will be of Hele-Shaw Martineau type. Electric power will be supplied at 220 volts by two 160-kw. Diesel-driven generators, running at 250 R. P. M., and one generator of 50-kw. capacity driven by a Diesel or semi-Diesel engine. For starting these engines a compressor set will be provided. It is intended that one large generating set will supply the electric power required at sea and normally in port. When all the winches are in operation, the power will be supplemented by the 50-kw. set. A small lighting set of 12-kw. capacity at 110 volts, with a semi-Diesel engine, will also be provided capable of being started by hand, and a rotary converter will be fitted. A special feature of the installation is the absence of high-pressure air compressors. The thrust blocks will be of the Mitchell type. The present intention is to carry out a series of test-bed trials of the main engines at the works before fitting them on board the ship.

Some particulars of the characteristics and broad principles which have influenced the design of the Scott-Still engine will be of interest. The engines consist of an ingenious combination of oil engines and steam engines. The main source of power is oil consumed within the cylinder for the down stroke, operating on the two-cycle principle, while steam forms a supplementary source of power, being used upon the up-stroke. The

oil fuel is injected into the cylinders without the use of high-pressure air. The engines are designed to reduce heat losses to a practical minimum, the primary consideration being to accomplish this in such a manner as to improve the thermal conditions of the working cylinders and so ensure the maximum efficiency from the fuel burnt therein. The combustion cylinder liners are of special form to secure immunity from heat cracks. On each cylinder top there is only one opening, and no separate cover is required for the combustion cylinder. Forming an integral part of each engine there is fitted a regenerator which is constructed like a small oil-fired water-tube boiler, and supplies steam for starting and maneuvering the engine. When the engine is started, the oil burners are shut off, and the regenerator then serves merely as a steam and water reservoir with which the water in the combustion cylinder jacket is in circuit. The heat from the jacket water, and that recovered from the exhaust gases through the medium of the regenerator and the feed heater incorporated therewith, all goes to form steam which performs useful work on the steam side of the engine pistons. The thermal working conditions of the cylinders result in the compression pressure being considerably less than in the ordinary Diesel type of engine. The valves on the steam side of the cylinders are operated by oil under pressure, which simplifies and facilitates the control of the engine for starting-up and while maneuvering. An impressive feature of the Scott-Still engine is the absence of the complicated array of rods, valves and cams associated with the ordinary Diesel engine. The results obtained from the experimental engine at Scotts' Works show that the Scott-Still engine has an advantage in fuel economy over the best Diesel engine; but even more important, from the shipowner's point of view, is the fact that the space occupied by such an installation and the weight of machinery in sea-going trim are considerably less than with Diesel engines of the same power. It will be seen that the Scott-Still engines have many novel and attractive features, and the results of the new Holt vessel will be awaited with the greatest interest.

The following is the full text of the report on the test of the "Still" engine carried out on October 5 and 6, 1921, by Captain H. Riall Sankey, C.B., C.B.E., R.E. (retired).

*The Object of the Test.*—The object of this test was to observe the general behavior and, more particularly, the fuel economy of a "Still" engine as far as could be done with an experimental unit.

The "Still" engine, in the design for marine purposes shown to me by Messrs. Scott, consists of groups of three cylinders working on a three-throw crankshaft; the top of each cylinder works as a two-stroke oil engine and the bottom as a steam engine. The steam is produced in a boiler by the heat transferred from the combustion part of the cylinder to the jacket cooling water and from the hot gases of the exhaust. The steam first enters the bottom of one of the cylinders which acts as an high-pressure cylinder and, after expansion, into the bottom of the remaining two which act as low-pressure cylinders. The steam is then condensed in the usual way, and is pumped back into the boiler through a feed-heater (heated by the exhaust gases) and the cylinder jackets.

The engine tested was one cylinder, such as would be combined into a group of three as above, the steam portion representing one of the low-pressure cylinders. The high-pressure cylinder of the group was improvised out of a single-acting steam engine working on an extension of the crankshaft of the main engine. Its cylinder was jacketed with exhaust

gases, which no doubt improved the economy of this cylinder, but not sufficiently to make it as economical as the high pressure cylinder of the group of three cylinders for reasons which will be given later. This auxiliary steam engine also added to the mechanical losses.

*Method of Testing.*—Many tests had previously been made with this experimental engine in accordance with a carefully-thought-out plan by a staff of observers who had become expert in taking the various readings. After due consideration I adopted the method of testing in use and thus my test may be regarded as concurrent with this series, and it may be said at once that it is in agreement with these previous tests. In support of this statement Fig. 1 is referred to, where it will be seen that the differences are well within the permissible errors in such tests.

The load was applied by means of a Froude hydraulic brake and the speed was regulated by hand. The fuel oil was measured by means of calibrated cylindrical tanks of small diameter, the readings being taken on external gauge glasses. Indicator diagrams were taken of the combustion cylinder, of the high-pressure and of the low-pressure cylinders. Pressures and temperatures were measured at numerous points, the latter by thermo-junctions and by mercury thermometers.

*Calibrations.*—The weights for the Froude brake were calibrated and its leverage checked. Readings of the oil measuring tanks were checked by weighing. The gauges were compared with a standard.

Two days, October 5 and 6, were occupied with this test; on the first day, full load, three-quarter load and half-load trials were made, and the second day was taken up with an overload trial and the various calibrations.

The duration of each trial was regulated so as to get a reliable measure of the fuel economy of the engine; its endurance had been already adequately tested on a previous occasion by a continuous 14 days' run at full load.

A peculiarity in testing this machine is that after a substantial change of load (say, from full to three-quarter), some little time must elapse before the temperature distribution, throughout the engine and the weight of steam settle down to the regime of the new load, and during this time no useful readings to obtain fuel economy can be taken.

The scavenge air was obtained by means of a blower, and after correcting for air blown away to waste to overcome certain surging difficulties due to this form of blower when employed to scavenge a single cylinder, it was found that the horsepower required was as follows:

| Overload. | Full. | Three-quarter. | Half. |
|---|---|---|---|
| 20.7 | 17.7 | 13.5 | 6.5 |

This horsepower has been deducted from the horsepower absorbed by the brake to obtain the effective horsepower given in Table I.

Throughout the test the engine ran perfectly and with very little noise; there was a marked absence of the jar usual in internal combustion engines—in fact, it ran like a steam engine.

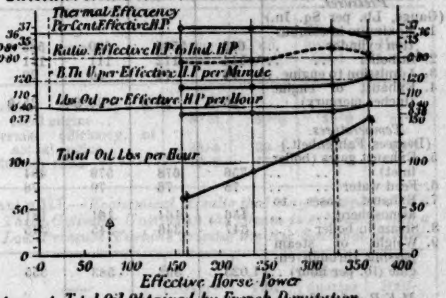
*Results of the Test.*—The essential results of the test are given in Table I below. In Fig. 1 are plotted on an "effective horsepower" basis the total oil consumption, the oil per effective horsepower per hour, the thermal efficiency, the British thermal units per horsepower per minute and the ratio of the effective horsepower to the indicated horsepower at the various loads. Also, by way of comparison, some of the results of the trials made on May 10, 1921, by the French engineers.

Attention is called to the following facts:

(*a*) The remarkable fuel economy and the corresponding high thermal efficiency.

(*b*) The small change in this economy from half to full load.
(*c*) The capability of considerable overload at a comparatively small increase in fuel consumption per effective horsepower.
(*d*) The high brake efficiency for a high compression oil engine.
(*e*) The low consumption of lubricating oil.
(*f*) The low temperature of the exhaust gases on final discharge.
(*g*) The entire absence of smoke, except a slight haze at overload.

**DIAGRAM GIVING THE PRINCIPAL RESULTS OF TABLE I.**


Thermal Efficiency
Per Cent Effective H.P.
Ratio Effective H.P to Ind. H.P.
B.Th.U. per Effective H.P. per Minute
Lbs. Oil per Effective H.P. per Hour
Total Oil Lbs. per Hour
Effective Horse Power
Total Oil Obtained by French Deputation

*Thermal Efficiency and Efficiency Ratio of Combustion Cylinder.*—At full load, the indicated horsepower developed in the combustion cylinder was 351.5, the oil consumption was 126.7 pounds per hour, and since the lower calorific value of the oil is 18,053 B. T. U. per pound, the thermal efficiency was 39.2 per cent. The thermal efficiencies for the other loads are given in Table II.

These thermal efficiencies have been compared with those of an ideal oil-engine working between the same temperature limits as the actual engine at the various loads; these temperature limits have been obtained by the use of a temperature-entropy chart, and the thermal efficiencies calculated, taking account of the variation of specific heat. The results obtained are given in Table II.

*Thermal Efficiency and Efficiency Ratio of Steam Cylinders.*—At overload, 1,025 pounds per hour of steam (saturated) passed through the high-pressure and low-pressure cylinders. In the case of the high-pressure cylinder the indicated horsepower developed was 22.8, corresponding to a feed of 45 pounds per indicated horsepower hour. The admission pressure was 111 pounds absolute per square inch, and the exhaust 22 pounds absolute per square inch, and for these steam conditions the feed of the Rankine engine is 21.5 pounds per indicated horsepower hour. Hence the efficiency ratio was 47.8 per cent, which is distinctively low for an high-pressure cylinder.

For the low-pressure cylinder the following are the corresponding figures:

Indicated horsepower developed, 38.9.
Feed, 35.5 pounds per indicated horsepower.
Rankine feed, 17.1 pounds per indicated horsepower.
Efficiency ratio, 48 per cent.

Usually the efficiency ratio of the low-pressure cylinder of a condensing engine does not exceed 40 per cent to 45 per cent owing, mainly, to the loss caused by "cutting off the toe of the diagram." The higher value of

TABLE I.

*Dimensions of Engine.*

| Main Cylinder : | | Auxiliary H.P. Cylinder : | |
|---|---|---|---|
| Stroke | .. 36 in. | Stroke | .. 22 in. |
| Bore | .. 22 in. diam. | Bore | .. 14 in. diam. |
| Piston rod | .. 6¼ in. diam. | | |

| | Over-Load. | Full Load. | Three-quarter Load. | Half-Load. |
|---|---|---|---|---|
| *Pressures.* (Gauge, Lb. per Sq. In.) | | | | |
| 1. Maximum in combustion cylinder | 615 | 595 | 600 | 585 |
| 2. In boiler | 114 | 112 | 111 | 111 |
| 3. Admission to engine | 96 | 78 | 50 | 29 |
| 4. Exhaust of engine (inches mercury) | 26·5 | 26·9 | 26·8 | 25·9 |
| *Temperatures.* (Degrees Fahrenheit.) | | | | |
| 5. Exhaust gases (boiler inlet) | 756 | 678 | 579 | 494 |
| 6. Feed water | 78 | 76 | 79 | 78 |
| 7. Exhaust gases to atmosphere | 156 | 164 | 161 | 181 |
| 8. Steam in boiler | 347 | 346 | 345 | 345 |
| 9. Weight of steam passing through engine (lb. per hour) | 1,025 | 892 | 583 | 355 |
| *M.E.P. referred to Combustion Cylinder.* | | | | |
| 10. Combustion | 89·53 | 81·37 | 71·27 | 56·63 |
| 11. Steam, high-pressure | 5·10 | 3·86 | 2·07 | 1·12 |
| 12. Steam, low-pressure | 6·46 | 6·14 | 4·75 | 2·72 |
| 13. Total | 101·1 | 91·4 | 78·1 | 60·5 |
| 14. R.P.M. | 129·8 | 125·0 | 111·3 | 97·4 |
| 15. I.H.P. combustion cylinder | 400·5 | 351·5 | 274·1 | 190·6 |
| 16. I.H.P. total | 451·7 | 394·7 | 300·3 | 203·5 |
| 17. Brake load (lb.) | 1,500 | 1,380 | 1,125 | 850 |
| 18. H.P. absorbed by brake | 387·9 | 345·0 | 250·4 | 165·6 |
| 19. Brake efficiency, per cent. | 85·9 | 87·4 | 83·4 | 81·4 |
| 20. H.P. used in scavenging | 20·7 | 17·7 | 13·5 | 6·5 |
| 21. Effective H.P. | 387·2 | 327·3 | 236·9 | 159·1 |
| 22. Ratio, effective H.P. to total I.H.P., per cent. | 81·3 | 82·9 | 78·9 | 78·2 |
| 23. Total oil consumption per hour, lb. | 147·2 | 126·7 | 90·9 | 61·0 |
| 24. Oil consumption per combustion I.H.P. per hour, lb. | 0·367 | 0·360 | 0·332 | 0·320 |
| 25. Oil consumption per effective H.P. per hour, lb. | 0·401 | 0·387 | 0·384 | 0·383 |
| 26. Thermal efficiency I.H.P., per cent. | 38·4 | 39·2 | 42·5 | 44·1 |
| 27. Thermal efficiency, effective H.P., per cent. | 35·2 | 36·4 | 36·7 | 36·8 |
| 28. B.Th.U. per minute per C.I.H.P. | 110·4 | 108·2 | 99·8 | 96·2 |
| 29. Ditto per effective H.P. | 120·6 | 116·4 | 115·5 | 115·2 |
| 30. Lubricating oil per hour, lb. | 0·6 | 0·6 | 0·6 | 0·6 |
| 31. Smoke | Slight haze | None | None | None |

40 per cent in this case is due to the heat transmitted from the combustion cylinder.

It cannot be doubted that in an actual marine engine of the type previously described where the high-pressure cylinder is at the bottom of a combustion cylinder, so that the steam receives heat during admission and expansion, the efficiency ratio of this high-pressure cylinder would be at least 90 per cent. As regards the low-pressure cylinder, an efficiency ratio

TABLE II.

| — | Over-Load. | Full Load. | Three-quarter Load. | Half-Load. |
|---|---|---|---|---|
| Thermal efficiency of ideal engine .. .. | 50·0 | 50·8 | 55·0 | 51·6 |
| Thermal efficiency of actual engine .. | 38·4 | 39·2 | 42·5 | 44·1 |
| Efficiency ratio .. .. | 76·8 | 77·2 | 83·3 | 85·4 |

TABLE III.—*Economical Results that may be expected in a Three-Cylinder Unit when the Steam is exhausted into a Low-Pressure Turbine driving the Air Scavenge Blower.*

| — | Over-load. | Full Load. | Three-quarter Load. | Half-load. |
|---|---|---|---|---|
| Total I.H.P. .. .. | 463 | 404 | 302 | 206 |
| Effective H.P. .. .. | 404 | 359 | 256 | 172 |
| Brake efficiency, per cent. | 87·2 | 88·8 | 84·6 | 83·2 |
| Oil per effective H.P. (lb. hour) .. .. | 0·364 | 0·353 | 0·355 | 0·355 |
| Thermal efficiency per effective H.P., per cent. .. .. .. | 38·7 | 39·9 | 39·7 | 39·7 |
| B.Th.U. per minute per effective H.P. .. | 109·6 | 106·3 | 106·8 | 106·8 |

of 90 per cent can be expected if the back pressure be increased to the release pressure by passing its exhaust into the low-pressure portion of a mixed pressure turbine provided for driving the scavenge air blower. This turbine should be supplied with a small amount of steam direct from the boiler to make up the power required by the blower, and for regulating purposes.

In the experimental engine tested a certain amount of the exhaust gases was used for jacketing the high-pressure cylinder (auxiliary engine); in the three-cylinder unit this waste of exhaust gases would not occur and a corresponding additional weight of steam would be produced in the boiler.

The mechanical losses of the auxiliary steam engine were included when determining the horsepower absorbed by the brake, but in a three-cylinder unit they would not occur. The auxiliary engine is single-acting and judging from general experience its mechanical efficiency at 40 pounds M. E. P. would be about 90 per cent, which works out to a loss of horsepower equivalent to 0.0342 × revolutions per minute. Being a single-acting engine, the loss is practically constant at all loads, and this horsepower can therefore be added to the effective horsepower.

The calculations to ascertain what improvement in economy may be expected, based on the above, are given in Table III.

January 25, 1922. H. RIALL SANKEY.

—"Engineering," February, 1922.

## NEW AMERICAN GASOLINE TURBINE.

Initial experiments of the development of an oil-turbine for motorships have recently been carried-out at Morton, Delaware County, Pa., by Otto Kories, a Diesel engineer employed at the Cramp shipyard, Philadelphia. The first unit has just run tests, but has been constructed under difficulties and not with the best of material due to the limited financial resources of the inventor. This is a small gasoline-turbine which later will be designed to run on heavy-oil. Early in February the first run was made. This lasted 15 minutes at 800 R. P. M., followed by 15 minutes at 1,000 R. P. M., 15 minutes at 1,200 R. P. M., 15 minutes at 1,500 R. P. M., all without load.

Then the engine was put on load and the speed reduced to 1,160 R. P. M., which showed the best results, ignition-air being at 102 pounds. It was connected to a 55-volt generator, which was operated at 232 ampères with ease, and the turbine run for one hour. It was then stopped and started several times with success. Afterward the turbine was dismantled and inspected, but no faults were found, only a few blades showing signs of having received a full blast of fire. About ¼ pint of lubricating oil was used for the bearings and two cams. Less than two quarts of gasoline was consumed on the entire run of 2 hours. The power output at full load was 17.1 B. H. P. For the one hour run at full load 1¼ quarts of gasoline, or 2.8 pounds, was used.

As the results of this run several minor changes are being made to the design. The compressor is of the two-stage rotary type, which compresses its air against the housing to 68 pounds at 1,000 R. P. M., 87 pounds at 1,200 R. P. M. and 111 pounds at 1,500 R. P. M. The inside diameter of the combustion-chamber is 3 inches by 1 inch width, the chamber being water jacketed. Piston-type valves, operated by cams, one of the latter serving two valves, are fitted, the valves being drilled horizontally through the center.

The operation is as follows: By slightly opening the gas lever and opening the starting-valve "full," air enters the turbine wheel, setting it in motion. At the same time the compressor blows the compressed-air through a specially designed atomizer, the mixture passing through the gas-intake manifold to the combustion-chamber via the mechanically operated valves. The valves then close the moment that combustion occurs, and the gases escape through ports and on to the turbine blades.—"Motorship," March, 1922.

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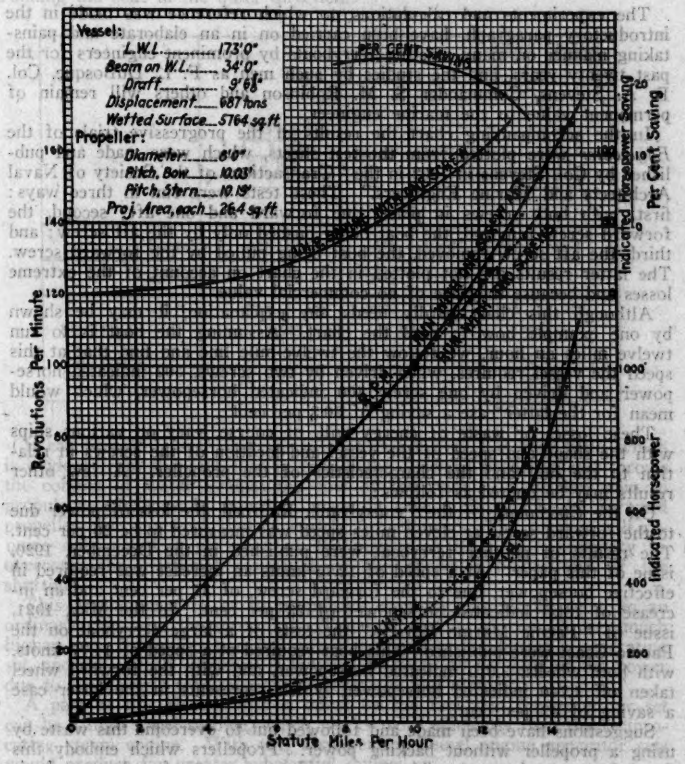
## FERRYBOATS CAN BE OPERATED ON MORE ECONOMIC BASIS.

### Outline of Method and Sketch of Arrangement of Machinery for Eliminating Waste on Double Ended Boats.

By H. Schreck.

The uneconomical propulsion of double ended boats led the writer to make a careful study of this condition, which finally resulted in a device which will eliminate the waste of power encountered on the present day arrangement of these boats. This problem has for years been a subject of calculations and tests in order to establish the amount of said waste of power.

This type of boat is in general laid out with one triple expansion engine or two compound engines in tandem arrangement in the middle body of the boat. The shaft runs to each end of the boat and carries one propeller at each end; one of these propellers pushing the vessel ahead and the other pulling the boat. When the boat is reversed the engine is reversed and in turn also the respective working of these two propellers. Some boats are driven by two through shafts in which case the above described plant is simply duplicated.




RESULTS OF PROGRESSIVE TRIALS OF FERRYBOAT "EDGEWATER."

Line — and points designate I. H. P. when boat is driven by screw astern only.

Line - - - and crosses designate I. H. P. when boat is driven by screw ahead and astern.

It is well known that the low propulsive efficiency of these boats is due to the dragging of the forward screw and to the hydrostatic pressure of the water column, which the forward screw builds up on the bow and on a great part of the forward length of the hull. It is readily seen that in this manner a large part of the energy which is put into the forward screw is very largely lost and the output of the engine must be increased by the amount of said increased resistance of the boat in comparison to an ordinary ship with one screw aft.

TESTS.

The experiments and calculations, to which reference was made in the introductory paragraph, have been carried on in an elaborate and painstaking manner on many existing ferryboats by prominent engineers for the past twenty years, and the studies by such men as F. L. DuBosque, Col. E. A. Stevens, Commander S. M. Robinson and others will remain of permanent value to the marine engineer.

In the accompanying chart the results of the progressive trials of the *Edgewater* were plotted from the test sheets, which were made and published by Col. Stevens in 1902 in the "Transactions of the Society of Naval Architects and Marine Engineers." These tests were run in three ways: first, with both screws in place, one forward and one aft; second, the forward screw removed, the boat being pushed only by the aft screw; and third, the aft screw removed, the boat being pulled by the forward screw. The latter results are not plotted in the chart on account of the extreme losses and because they are of no commercial value.

Although this chart hardly needs any explanation, it may be shown by one example how to read the chart. Assuming the boat is to run twelve miles an hour, we follow the twelve-mile line and find that at this speed the vessel requires, when driven by two screws, 502 indicated horsepower and driven by one screw 400 indicated horsepower, which would mean for the latter case a saving of 20.3 per cent.

These losses or waste of energy will, of course, vary on various ships with the shape and speed of the vessel and location of the screws in relation to the hull and the characteristics of the propeller. A few other results may be quoted as follows:

On the *Cincinnati*, of the Pennsylvania Railroad, the loss of power due to the forward screw at eleven miles speed was measured to be 20 per cent. The results of another ferryboat were published in the December, 1920, issue of this paper. At a speed of 12.75 knots an increase was required in effective horsepower, due to the forward screw, of 37 per cent and an increase of total indicated horsepower of 23 per cent. In the May, 1921, issue of "Pacific Marine Review," the tests of a large ferryboat on the Pacific Coast were published. This boat required at a speed of 14.77 knots, with both wheels, 3,010 indicated horsepower, and with the forward wheel taken off 1,740 indicated horsepower, which represents in the latter case a saving of 42 per cent.

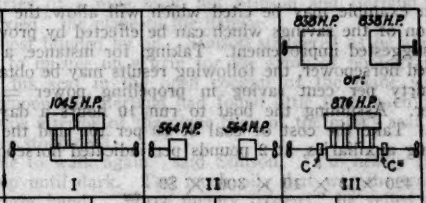
Suggestions have been made and followed out to overcome this waste by using a propeller without backing power. Propellers which embody this demand, as much as possible, are running on some ferryboats. It is claimed, however, that the maneuvering of the boat is very unsatisfactory

TREND OF ELIMINATING THIS WASTE.

There are three characteristic methods of propulsion of double ended ferryboats and in illustration some of the figures of a previously mentioned publication by Commander Robinson will be utilized.

If we consider his boat of 191 feet length, 44 feet beam, and running at a speed of 10 knots, the total towing power with two screws of 680 effective horsepower and with only the after screw 496 effective horsepower the following three arrangements may be investigated for this boat:

1. The present day arrangement, that is, one main engine.
2. Two motors, each transmitting about half of the power. However, each one can be turned at its respective most favorable speed.
3. Two motors, one at a time transmitting the full power for pushing the boat, while the other motor turns the screw over at about the speed of the boat, but without doing any work. This condition is reversed for running the boat in the other direction.




| Screw | Aft | For'd | Aft | For'd | Aft | For'd |
|---|---|---|---|---|---|---|
| R.P.M. | 116.9 | | 114.7 | 119.8 | 131 | 117 |
| I.H.P. | 656 | 389 | 564 | 460 | 838 | 38 |
| I.H.P. Total | 1045 | | 1024 | | 876 | |
| Gain over I | | | | | 16% | |
| Gain over II | | | | | 14.5% | |

*Clutch

It can be readily seen that the gain of case 2 over 1 is very little, which is evident from the fact that in case 2 the forward propeller still throws the column of water against the bow of the vessel. A marked gain in propulsive efficiency is, however, made in case 3, in which the forward propeller does no work, but is merely turned over. The turning will require only about 4 per cent of the power, as it is only necessary to overcome the frictional resistance of the propeller blades and of the shafting.

This arrangement is, as to propulsive economy, practically equal to an ordinary ship with the screw aft. The first cost is, of course, very much higher since it requires two full power motors and in addition a generating set, and the overall efficiency is lowered by 15 to 18 per cent of the power due to the electric transmission by generator and motor.

A part of this electric loss is offset by the fact that the generator set can be a high speed turbine, which works more efficiently than the reciprocating steam engine. With this arrangement Diesel engine generator sets would produce far greater substantial saving.

### SUGGESTION OF THE WRITER.

In order to accomplish the maximum propulsive efficiency, the writer suggests an arrangement which will combine the advantages of cases 1 and 3 at the smallest cost of installation and the possibility of building it in on existing boats (a patent on this combination is pending).

The suggestion is to interpose a clutch at each end of the main propelling unit between same and the two propeller shafts. This clutch, which may be a magnetic or hydraulic one, will allow varying the power transmitted and the speed of the after shaft in relation to the forward shaft; that is, when the boat is going ahead almost the full power of the main engine will be transmitted to the aft screw, while the forward screw will be merely turned over at about the speed of the boat, but without doing any propulsive work. Clutches that fulfill the above requirements have been investigated and are being built.

EXPECTED SAVINGS.

An example may be cited which will allow the reader to form a conception of the savings which can be effected by providing a ferryboat with the suggested improvement. Taking, for instance, a ferryboat of 1,400 indicated horsepower, the following results may be obtained:

Thirty per cent saving in propelling power = 420 indicated horsepower. Assuming the boat to run 10 hours a day for 300 days of the year. Take the cost of coal at $9 per ton and the coal consumption, including auxiliaries, at 2 pounds per indicated horsepower.

| | |
|---|---|
| $\frac{420 \times 2 \times 10 \times 300 \times \$9}{2{,}240}$ = | $10,120 |
| Reduced cost in handling of coal and reduced boiler repair bill, estimated | 3,500 |
| Saving per year | $13,620 |

It will be noticed that the estimates are made very conservatively. The saving of 30 per cent is taken as an average of actual tests. Also 10 hours' operation is very little, since most ferries run for 14 to 16 hours and some of them for 24 hours. Very conservative also is the figure of 2 pounds of coal per indicated horsepower for compound engines, but the writer has tried to make all his statements as carefully as possible.—"Marine Engineering and Shipping Age," March, 1922.

## LARSEN PLANE ESTABLISHES WORLD'S ENDURANCE RECORD.

Flying continuously for 26 hours, 19 minutes and 33 seconds in gales, snow and a zero temperature, Edward Stinson and Lloyd Bertaud in a Larsen all-metal monoplane at Roosevelt Field, near Mineola, L. I., set a new world's record December 30, for sustained flight in a heavier-than-air machine. Averaging 85 miles an hour in circling above the flying field and surrounding country, the monoplane traveled at least 2,200 miles, or, translated into land distance, the equivalent of the Pacific gap between California and Hawaii.

Had the plane flown straightaway the pilots would have landed in Reno, Nev. The feat was all the more remarkable when it is compared with the previous endurance record of 24 hours, 19 minutes and 2 seconds made in France by Lucien Boussoutroy and Jean Bernard. Their flight was made on a clear day in June of last year after careful preparation. Stinson and Bertaud agreed to make the attempt for J. L. Larsen, designer of the plane, the day before the start of the flight.

Numb with cold and exhausted by lack of sleep, Stinson and Bertaud brought the monoplane down to earth at 11:17:50 o'clock A. M., December 30. They both maintained that they would have remained up several hours longer had they not been forced to descend by reason of a faulty oil connection with which they had been struggling since 3 o'clock in the morning.

TAXIED BACK TO HANGAR.

When the J. L.-6, landed in the middle of Roosevelt Field within a few feet from where it took off the day before, Mr. Larsen, who had maintained a constant vigil, started out in his automobile to bring the two aviators to the hangars a quarter of a mile back, but Stinson and Bertaud stuck to their posts and "taxied" the plane to the hangar.

Both fliers were almost blinded by oil. Stinson staggered as he alighted, but Bertaud was able to muster up a grin for the army of photographers. Stinson had not left his seat in the open cockpit of the plane since 4 o'clock Thursday afternoon and Bertaud had found little time to enjoy the comfort of the limousine interior of the plane back of the pilot's seat. For the time both men were stone deaf.

"It's good to be able to walk again," said Stinson. "I'm nearly frozen. We could have said up until dark. Too much oil; that was the trouble." Stinson held up his right hand. "Three fingers frozen!" he yelled at the top of his lungs. "One finger frozen on the other hand, too. I'm going home to bed. I feel as if I would never get warm again."

Bertaud fared better, but he agreed that the cold had been "terrific." He, too, expressed a desire to sleep, though he related his experience before leaving the field. Mrs. Stinson was on hand to greet her husband, and brought sandwiches, but "Eddie," as he is known to hundreds of army aviators whom he trained during the war, was too tired to eat. He left at once for his home in Hempstead. Describing the flight to newspaper men, Bertaud said:

"Our first four hours in the air yesterday were flown between fifty and seventy feet above the ground. The snowstorm raging then was so heavy that we could not go any higher and see the earth. We had to dodge church spires and chimneys frequently. Once when I was piloting the machine we almost hit the old chimney of the Camp Mills Hospital.

"Stinson and I did not do much talking until after we found we had been flying over twenty-five hours and knew that we had beaten the world's record. We both turned around in our seats and shook hands with each other.

"We encountered our first difficulty about 2 o'clock this morning, when the oil from the auxiliary tank began to thicken from the cold and would not feed well. We discussed the question as to whether we should land. We decided not to quit as long as the plane would remain in the air. We hit upon the plan of pouring the oil into the tank and not depending on the feed pipe.

POURED OIL IN TANKS.

"While one of us steered the other had to lift the five-gallon auxiliary oil cans, into which we had punched holes, in order to pour the heavy mixture into the tank. It was so thick that we could pour only about a quart at a time. Great care was required to keep the oil from running out, thereby burning out the bearings and destroying the motor. Incidentally, our hands, which had become numb just before midnight, got back some of their feeling with this strenuous work.

"Flying around and around the country was monotonous, especially after darkness set in. The fatigue from the cold was terrible. We did not dare to go to sleep for fear that we would become so chilled and numbed that when we awoke we would not be able to handle the plane. We had to maintain a constant vigil to keep the plane at an altitude where it would consume the least amount of gasoline.

"When we started on the flight the motor was turning up 1,400 revolutions per minute and when we landed it was 1,075 per minute. The motor under ordinary circumstances will consume about seventeen gallons of gasoline an hour. With the engine throttled down we used about twelve. The principal factors in a test of this kind are reliability, economy and good handling. During the night we never flew out of sight of the night flares at Roosevelt Field and Larsen Field, at Central Park, L. I. Sometimes during the long hours of the night we hovered over Freeport to vary the monotony.

"I have never flown more than twelve continuous hours before and I doubt if Stinson has. I do not care to try it again under such adverse weather conditions as we had on this occasion."

ATE NOTHING IN FLIGHT.

Bertaud said that neither he nor Stinson had eaten any of the food that they had taken along with them on the trip. They each drank a little hot coffee during the flight, he said. The J. L.-6 "took off" from Roosevelt Field at 8:58:15 on December 29 in a driving snowstorm. The two pilots took no particular precautions to protect themselves against the blustery weather, depending for warmth while in the air on a special heating arrangement in the cockpit, which led into the spacious interior compartment which had been fitted up comfortably with a bed.

Stinson wore a sweater and flying coat over a khaki uniform, while Bertaud was attired in a civilian suit and leather coat. The plane carried 2,200 pounds of gasoline and a gross load of about 5,300 pounds.

Seven judges appointed by the Contest Committee of the Aero Club of America acted as starters and watchers of the flight, working throughout the day and night in two-hour shifts. The judges were H. G. Scholler, W. M. Royse, H. G. Davison, L. Balch, R. B. Uber, Earl Holtham and H. Sushan.

In a shack heated by three oil stoves the judges watched the plane perform its endless encirclings. At nightfall three flares were lighted and kept burning until daylight to guide the airmen. At 9:20 yesterday morning the aviators had broken the world's record, but Stinson and Bertaud gave no indication of coming down. From that moment on it was simply an attempt to hang up a new endurance record.

AMERICAN MADE THROUGHOUT.

J. L. Larsen, owner and designer of the monoplane, who has flown more than 300,000 miles in this country and Europe in J. L.-6's, said that in a sister plane of the one used yesterday he had traveled by air from the southern part of Mexico to within two hours' air distance of the Arctic Ocean and from Boston to San Francisco. "And the plane is all American made," he said. "Made in America by American workmen and of American material. I am happy to know that I have been able to capture another aeronautical record for the United States."

Mr. Larsen said he flew the record winner recently to the air meets at Kansas City and Omaha. The plane, constructed of duralumin throughout, has a wing span of fifty feet and a length over all of thirty-one feet. It

stands eleven feet high. It is motored with a 185-horsepower B. M. E. engine and has an enclosed cabin seating six persons. Major Gen. Mason Patrick, Chief of the United States Air Service, sent a congratulatory message to Stinson last night.

"The Army Air Service," he said, "is pleased to extend to a former member its congratulations on the remarkable duration record just attained by you."

Augustus Post, Executive Secretary of the Aero Club of America, said the feat was but another demonstration of the possibilities of the aeroplane as a carrier of commerce. One did not have to look very far into the future to picture non-stop flights between New York and Chicago and other cities as a common every day occurrence.—"Aerial Age Weekly," January, 1922.

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## THE VACUUM AIRSHIP.

Although the airship in this country is for the time being virtually dead, activity in connection with it is still being displayed in several foreign quarters. There are, of course, some here who even yet have not abandoned their faith in its future, but it cannot be denied that very little impetus now remains in this country behind the development of airships. They have been virtually discarded for naval and military purposes; those which we own have been the subjects of frequent mishaps, culminating in the lamentable disaster to R-38, and all efforts on the part of the Government to induce anyone to take over and operate the existing fleet on a commercial basis have failed. Abroad the discouraging factors have apparently been less acute or less acutely felt. Certain it is that here and there people are to be found who are laboring undauntedly to improve the design and construction of airships. In the United States, for example, full-sized experiments are being conducted by the Navy Department on the use of helium or a helium mixture as a substitute for hydrogen. But of all the efforts now being made none exceeds in scientific interest and fundamental significance those in connection with the Vaugean airship at present in course of construction at Milan. This vessel, of which a description was published recently in "Le Génie Civil," does not derive its buoyancy either from hydrogen or helium. Instead, M. Vaugean seeks to secure flotation by exhausting the air from a metal-sheathed envelope. Such a proposal is, of course, far from being new. Indeed, the use of rarified air as represented by Montgolfier's fire balloon of 1783 actually takes precedence of hydrogen or coal gas in point of time. It has, too, so far failed to yield practical results of any importance, and in modern times has been shown mathematically to be impossible of application to airships of the self-propelled type. M. Vaugean holds, however, that the mathematicians' conclusions are premature, and that recent metallurgical discoveries combined with the adoption of certain principles of construction and working which he has devised, bring the proposal inside the reach of practical attainment.

The rock on which the vacuous balloon or airship has hitherto foundered is the difficulty or impossibility of constructing the envelope of sufficient size to give the required lift and of sufficient strength to resist the air pressure. A cubic foot of air weighs just over fourteen times as much as a cubic foot of hydrogen at the same temperature and pressure. It therefore follows that a given volume filled with hydrogen at atmospheric pressure is equivalent in lift to the same volume exhausted of air to a pressure very slightly in excess of one pound per square inch absolute. The outer skin of the equivalent vacuous airship or balloon is thus exposed at

the ground level to a resultant pressure of 13.7 pounds per square inch, a figure small, of course, when contrasted with the pressure inside a boiler or outside a tube, but of prohibitive magnitude when applied to the exterior of a vessel of the size and construction of an airship or balloon. The essential characteristic of M. Vaugean's scheme is that he does not propose to employ such a degree of rarefaction *at the ground level*. He designs his airship to travel, let us say, at an altitude of 10,000 feet. At that height the atmospheric pressure is about 10 pounds, so that with an internal vacuum of one pound absolute the envelope will be subjected to a resultant pressure of 9 pounds per square inch, a value which, he believes, he can design it to withstand. To descend he increases the internal pressure in such a way that when the ground level is reached it amounts to 5.7 pounds—that is to say, in such a way that the resultant pressure on the exterior of the envelope remains constant at 9 pounds throughout the descent. It is to be noted that were the internal pressure not increased and were the vessel pulled down to the ground level by means of ropes, it would arrive at the surface with a considerable amount of surplus lift. By increasing the internal pressure in the manner stated the lift remains constantly equal to the weight of the structure throughout the descent. Such, in brief, is the essence of M. Vaugean's scheme. As for practical details, it will perhaps be sufficient to say that it is proposed to use aluminum-copper-manganese alloys for the structure and sheathing of the envelope, to maintain and vary the internal vacuum by means of pumps assisted by the heat of the exhaust from the engines, and to carry the vacuum in three or more concentric chambers, each, passing inwards, containing a higher degree of rarefaction than the next outer one. By the judicious choice of the temperatures at which the contents of the three concentric chambers are preserved M. Vaugean estimates that it will be possible to operate the vessel with an effective pressure of only 2.1 pounds per square inch on the outer skin, 3.5 pounds on the intermediate, and 4.5 pounds on the inner.

The scheme is doubtlessly open to criticism at several points. For instance, it appears certain that it is bound to result in an increase in the dimensions of an airship designed to give a specified lift. That such will be the case can readily be shown for the vacuous airship without concentric chambers. Taking the internal pressure at the ground level at the value previously mentioned—namely, 5.7 pounds—a simple calculation shows that to develop the same lift as a hydrogen airship the vacuous vessel would have to be 1.55 times as great. Such a magnification of the dimensions of a type of vessel already objectionably large cannot be regarded with approval, but it is not of itself a fatal disadvantage. From the practical rather than the theoretical point of view criticism of the scheme will no doubt be chiefly directed towards the difficulty of creating and maintaining a high degree of vacuum in a vessel the capacity of which may be anything from one to three million cubic feet. The effort required to maintain a good vacuum in a condenser is well known, but the two cases are, we think, hardly comparable, although we feel certain that this question of maintaining the vacuum will be quite the most difficult that M. Vaugean will have to face. It should be remembered, however, that by carrying the vacuum in three or more concentric chambers differentially exhausted, not only is the crushing stress reduced, but the tendency to lose the vacuum by leakage is favorably modified. In addition, the arrangement will no doubt help the vessel in the matter of the size of the exhausting plant to be carried on board, for operational changes of the internal pressure would no doubt be effected in but one of the chambers, leaving the others under the pre-arranged vacuum. The

whole scheme is very highly interesting. It may not be completely practicable at the present moment, but it might well become so with a little experience or, at the worst, as a result of some apparently small advance in the metallurgy or in some other direction. It is at any rate worthy of scientific attention, for if it is or becomes practicable it will do very much more for the airship than would the discovery of a cheap, plentiful and handy source of helium or other light non-inflammable gas.—"The Engineer," January, 1922.

# BOOK REVIEWS.

THE ENGINEER SERVICE IN WAR. A Manual of Instruction. (The Society of American Military Engineers.)

A very concise and comprehensive description of the duties of an engineer in time of war. It should be especially useful to engineer officers in the Reserve Corps of the Army.

STANDARD SEAMANSHIP FOR THE MERCHANT SERVICE. By Felix Riesenberg, C. E. (D. Van Nostrand Co.). 625 illustrations, numerous tables.

An excellent treatise on Seamanship. When Seamanship is just spoken of it generally brings to mind the art of handling ships, whether sail or steam, knotting and splicing, Rules of the Road, etc. This publication goes farther in that it gives the reader an idea of the structural work necessary in ship construction, tables of weight and strength of ropes, both Manila and wire, working loads for blocks and many others which are of value. It is impossible to bring out all the many points touched upon which are of interest and practical use.

The Merchant Service Officers will find in this book a valuable aid in studying the subject of Seamanship, handling and stowage of cargo and other problems which are ever present in their service.

## NOTE.

GEORGE MONTEFIORE FOUNDATION. Triennial Prize.

ARTICLE I.—A prize, the amount of which is the accumulated interest of a capital of 150,000 francs, Belgian, at three per cent, is granted every three years as the result of an international competition for the best original work presented on scientific advancement and progress in the technical applications of electricity in all its branches, exclusive of works of popularization or simple compilation.

ARTICLE II.—The prize bears the name of George Montefiore Foundation.

ARTICLE III.—Only works presented during the three years preceding the meeting of the jury are admitted to the competition. They must be in French or in English and may be printed or in manuscript. However, the manuscripts must be typed and, in any case, the jury can decide as to the printing.

ARTICLE IV.—The jury is composed of ten electrical engineers, five Belgian and five foreign, under the presidency of the professor-director of the Montefiore Electrotechnical Institute who is by right one of the Belgian delegates. With the exceptions stipulated by the founder the latter must be chosen from those holding diplomas from the Montefiore Electrotechnical Institute.

ARTICLE V.—By a majority of four-fifths in each of the two sections, foreign and national (which must to this end vote separately), the prize may be exceptionally divided.

By the same majority the jury can grant a third of the available funds, at most, for a capital discovery to a person who did not take part in the contest or for a work which, without being completely within the program, shows a new idea capable of having important developments in the domain of electricity.

ARTICLE VI.—In case the prize is not given or if the jury awards only a partial prize, the whole sum made thus available is added to the prize of the following triennial period.

ARTICLE VII.—Typewritten works may be signed or anonymous. Any work will be considered anonymous that is not provided with a legible signature and with the complete address of the author.

Anonymous works must bear a device, repeated on the outside of a sealed envelope attached to the enclosure; inside this envelope must be legibly written the name, Christian name, signature and residence of the author.

ARTICLE VIII.—All works whether printed or typed must be submitted in twelve copies; they must be addressed postpaid to the Secretary-Archivist of the George Montefiore Foundation, at the headquarters of the Association, "rue Saint-Gilles, 31 Liége (Belgium)."

The secretary-archivist will acknowledge the receipt of enclosures to authors or contributors who make themselves known.

ARTICLE IX.—The works which the jury decides to print will be published in the "Bulletin de l'Association des Ingénieur électriciens sortis de l'Institut électrotechnique Montefiore." For this publication there is no charge made to the authors nor any rights in their favor. They are allowed nevertheless, as a matter of courtesy, twenty-five copies.

For this publication the English tests can be translated into French by the Association.

THE 1920 COMPETITION CHANGED BY EXCEPTION TO 1923.

The amount of the prize to be awarded is twenty-one thousand francs.

The last date for the acceptance of works to be submitted to the jury is April 30, 1923.

The works presented must bear at the top the caption: "Work submitted in the competition of the George Montefiore Foundation, 1920 session (1923)."

*For the Administrative Council of the Association of Electrical Engineer Graduates of the Montefiore Electrotechnical Institute:*

*The Secretary General,* L. CALMEAU. *The President,* OMER DE BAST.

## ASSOCIATION NOTES.

The Society held its annual banquet on March 18th at the New Willard Hotel, Washington, D. C. There were present one hundred and eighty-six members and guests.

The event was so successful that the Council rendered a vote of thanks to the banquet committee, composed of Commander F. J. Cleary, U. S. N.; Commander H. S. Howard (CC), U. S. N.; Commander H. A. Stuart, U. S. N., and Lieutenant M. R. Daniels, U. S. C. G. Commander Cleary acted as Toastmaster and the very able manner in which he conducted the program of the evening added materially to the enjoyment of the occasion.

Several distinguished members of the Cabinet, of the Congress and of the Naval Service were guests. The Society was particularly fortunate in being able to present a number of intensely interesting speakers, as follows:

"The Navy,"
THE SECRETARY OF THE NAVY.

"The Treaty Limiting Naval Armaments,"
SENATOR L. HEISLER BALL.

"The Arms Conference,"
THE ASSISTANT SECRETARY OF THE NAVY.

"The Congress and the Navy,"
REPRESENTATIVE LEMUEL P. PADGETT.

"Personnel Necessary for the Navy as Limited by the Arms Conference,"
ADMIRAL R. E. COONTZ, U. S. N.

"The Influence of the Navy on the Steel and Shipbuilding Industries of the United States,"
MR. HOMER L. FERGUSON.

"Motive Power as a Factor in Winning Naval Actions,"
REAR ADMIRAL J. K. ROBISON, U. S. N.

REAR ADMIRAL R. S. GRIFFIN, U. S. N.,
Retired, former Engineer-in-Chief of the Navy, also delivered an informal address.